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PRODUCT ENGINEERING

MAY
1934

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Types and Uses of Resistance Welds
Locking Bolts—Magnetic Contactors
Defects in Single-Bearing Mountings
Stress Concentration—Jewel Bearings

•

Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.

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IN THIS ISSUE

Resistance Welding—After explaining the scope and limitations of the six most common types of resistance welding the author shows how they may be applied by the designing engineer.

Bearings—How to design jewel bearings, is the subject of one article, and how not to apply sleeve-type bearings is the text of another article. Together they cover the field of bearings from instruments to machinery.

Stress Concentration—It seems strange that the strength of a member can be increased by cutting out some material. Dr. R. V. Baud tells how.

Electric Motors and Controls—Woodworking machinery embodies the most advanced applications of motors and controls as shown in this issue. And the article by E. W. Seeger explains the characteristics of various types of contacts and magnets for magnetic contactors.

COMING ISSUES

We have received numerous requests for articles that will give specific data on the design of bearings, bearing materials, types of cylindrical bearings and bearing lubrication. In the June number we begin a series that will cover all these subjects, written by C. H. Leis, chief engineer of the Johnson Bronze Company.

Many readers will remember the article by F. L. Haushalter on rubber for isolating vibrations. He has written another article for *Product Engineering* on rubber for impact and abrasion, which will appear in the June number.

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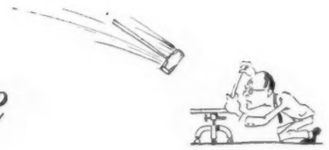
Intimate Correspondence



Dry Bearings

PRODUCT ENGINEERING
NEW YORK

What are the chances of our getting something on lubrication fairly soon? I have been a subscriber to *Product Engineering* since the first issue and have seen nothing yet on this subject.

All these machines surely do not run dry.

H. J. WILLIAMS,
McKeesport, Pa.

THEY certainly do not run dry, and there certainly have been articles on lubrication, but not so many recently. To remedy this deficiency the chief engineer of one of the leading manufacturers of plain bearings is now preparing a special series of articles on this subject which will include plenty of basic information.

In addition research and service engineers of several oil companies and the engineers of several companies making anti-friction bearings are contributing material on the subject of lubricating anti-friction bearings.

Addresses

PRODUCT ENGINEERING
NEW YORK

Under the heading "Developments in Product Design" many new devices are mentioned and the names of the manufacturers are given but not their addresses. I wish you would include this in the future so that one may be able to write direct for information or data.

J. W. MORTON,
Philadelphia, Pa.

IN the department devoted to product developments we have felt that the brief description of the engineering features of the development was the important thing rather than the name and address of the manufacturer. Items in this department are published to keep readers of *Product Engineering* abreast of developments in all of the various branches of engineering design in our field. In most cases the products described are not ones that would ordinarily be purchased by *Product Engineering* readers. That is where they differ from the items published in the new parts and materials department which is devoted to innovations that naturally would be purchased by *Product Engineering* subscribers. If other readers feel as Mr. Morton does we shall be glad to include the appropriate addresses. Otherwise we shall simply save the space and answer requests by letter.

West Coast

PRODUCT ENGINEERING
NEW YORK

Certainly do get benefit from reading *Product Engineering* as it lets us fellows on this Pacific Coast know what's going on in the East in the design business.

Why don't you get some material in your paper relative to West Coast activities along design lines? Although there is not much manufacturing in the Northwest where I am located there is plenty down around San Francisco and Los Angeles, and I am sure there are some good writers who could get this dope in shape for you down there.

H. A. PEARSE,
Seattle, Washington

SOME of the best articles we have had have come from the Pacific Coast. For instance, Mr. Griffith's article on springs which received many comments from readers of *Product Engineering*. We are not in the least particular where good articles come from and if they are good they are just as welcome from Timbuctoo as they would be from Toledo.

From England

PRODUCT ENGINEERING
NEW YORK

On page 2 of the September, 1933, issue of *Product Engineering* Mr. Wastenbarger of the Hocking Glass Co. asked you in a letter whether you could supply him with copies of "Engineering Standards for the Design of Springs" by W. M. Griffith which apparently appeared in the July Number of this publication. We should be very grateful if you could supply us with one copy also, or if this is not possible inform us from where a copy could be obtained.

L. D. WILLIAMSON,
Amherst Villiers Airmotors

SPEAKING of West Coast articles, and particularly the one by Mr. Griffith, here is evidence of its value and popularity, also of enduring quality and the value of this page as a forum.

Motor Inquiry Form

If not asking too much we would be glad to receive a dozen additional forms.

. . .

If you can spare us a half dozen more of these sheets we would thank you for same.

. . .

We can use at least 100 copies of inquiry form for electric motors.

. . .

We would be pleased to have a dozen copies as per your offer.

. . .

If it is not too much trouble we would appreciate your sending us 200 additional copies of this form.

. . .

We wonder if you would be kind enough to send us a quantity of these forms.

. . .

We can use 500 copies of the reprint of your motor inquiry form.

. . .

We would be interested in receiving a quotation on 1,000 reprints of page 145 covering the questionnaire, and if the reprints could include page 144, which is the accompanying article, we would sooner have them that way.

THESE are just a few of the requests for copies of the inquiry form for electric motors published in April *Product Engineering*. We meant what we said when we offered to supply additional copies with our compliments. If you can use some don't hesitate to write for them, and for as many as you want. There is no limit.

Special Appreciation

PRODUCT ENGINEERING
NEW YORK

I am a new subscriber to your magazine and wish to express my special appreciation for the charts on "Properties of Phenolic Laminated Molded Materials" (Dec., 1933).

This way of presenting facts is, in my estimation, especially useful to one who is not directly connected with the subject.

I hope you will continue to put in similar charts, from time to time.

R. C. HITCHCOCK,
Westinghouse Electric & Mfg. Co.

WE could not resist putting this one in because we like bouquets as well as anyone else.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MAY, 1934

McGRAW-HILL PUBLISHING CO., INC. VOLUME FIVE, NUMBER FIVE

A Job For Engineering Societies to Start

IT IS only human to choose for a new material a name that is descriptive of the particular application for which it was developed. That this type of nomenclature is neither sensible nor practical is proved by the confusion, ambiguities and absurdities that have been brought about by the general acceptance of such names. Gun iron is no longer used for cannons. Drill rod, originally a high carbon round steel rod ground to close tolerances, is now also the name for such rods made of special alloy steels, and the same name is also being applied to square rods ground to accurate dimensions.

It is not possible to legislate against the use of a meaningless vocabulary. But in the engineering profession there are organizations which by common consent have been entrusted with the job of bringing order out of chaos. Both the Society of Automotive Engineers and the American Society for Testing Materials have accomplished much relative to standardization of materials. The excellence of the S.A.E. system of numbering alloy steels is attested by its wide acceptance even outside the automotive field.

In the A.S.T.M. Standards we find such names as alloy tool steel, marine boiler steel plate, structural rivet steel, cartridge brass, ferrule stock, valve bronze and staybolt iron. In fact, it appears to be the aim of the A.S.T.M. to build specifications and name the materials according to their use. Already this is resulting in confusion and dissatisfaction, which is bound to grow worse as time goes on.

Here is an opportunity for our major engineering societies to render valuable service. Let them get together and formulate a logical and sensible nomenclature for materials. A good start has been made on steels by the S.A.E. Why not continue the good work? Such an accomplishment would benefit not only the "tonnage users" of materials, but the little fellows who also pay dues.

The American Society of Mechanical Engineers, which up to this time has done more with dimensional standards than with standards for materials, might well give attention to this vexing problem. Indeed, being one of the major societies and directly in the field concerned, it is the logical one to initiate such a movement, and the American Standards Association seems to be the proper organization to carry on the job.

Try Dynamite

AN INDEPENDENT designer observed the other day that no improvement in railroad car or locomotive design, at all radical, has ever come out of a car or locomotive building company.

Admitting that this observation is an overstatement, most engineers will agree that there is some ground for it, and that it applies in some measure to their own companies, no matter what the product. If they are honest they will take some of the responsibility. But they will also rise up in their own defense and point out how their initiative is stifled by unprogressive sales and production heads.

Many a promising technical development, entirely sound from the engineering point of view, has been squelched by a stand-pat production staff that refused to face the replacement of existing production facilities unequal to the task of making anything other than the old-line product. Or by a stick-in-the-mud sales department obsessed with the belief that price is the only weapon.

In fairness to harassed production and sales departments let us admit that four years of depression operating and depression selling have unquestionably left their mark. But, accepting this explanation, isn't there all the more reason for seeking the kind of dynamite needed to clear the path for technical and design advances? Perhaps it lies in giving the engineering department more of a say where innovations are concerned.

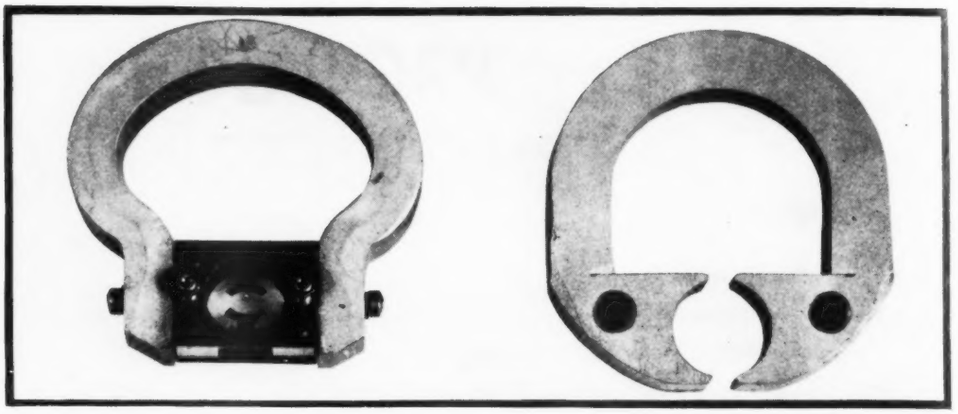


Fig. 1—Three-piece permanent magnet assembled by welding (right) replaces the 21-piece machine-screw assembly on the left. The welded design also eliminates annealing and drilling operations in the hardened steel material

Types of Resistance Welds And How They Are Used

F. S. STICKNEY
Meter Engineer
Westinghouse Electric & Mfg. Company

SELECTION of resistance welding rather than some other means of fastening is determined by consideration of the following factors: (1) Necessity of disassembly; (2) Mechanical strength required; (3) Kind of material used; (4) Cost of operation; (5) Cost of preparing parts; (6) Equipment available.

Since a welded assembly cannot be disassembled without destruction, resistance welding, rather than bolted or riveted fastenings cannot be used where the assembly must be such that it can be taken apart readily. But where permanence is desirable, a welded fastening is more rigid than a machine screw assembly and will not be affected by vibration. Particularly for apparatus intended to be of sealed or tamper-proof construction, such as the watthour meter cover shown in Fig 2, resistance welding has an advantage over a riveted fastening. Rivets could be removed and replaced, but pieces welded on cannot be detached without irreparably destroying the assembly.

To permanently attach the pole-pieces to the permanent magnet assembly shown in Fig. 1, the welded assembly shown at the right not only elminates 18 parts and a number of drilling and tapping operations used in the machine screw assembly, but also provides a greater amount of structural rigidity.

Greater strength can be obtained in welded joints than bolted, riveted, soldered or brazed fastenings. When the entire areas in contact are fused, the resistance welded fastening, like fusion welding, develops the full strength of the parts. The weld itself is stronger than the surrounding metal, failure occurring by the welded portion tearing away from adjacent metal. Welding has the further advantage over riveted or bolted fastenings in that vibration or extreme temperature variations cannot cause the welded connection to become loose.

Metals having moderately high electrical resistivity and low thermal conductivity are most readily welded. For example, iron and steel weld uniformly with a moderate amount of current, while copper and aluminum require a high current and careful adjustment of the welding machine. Between these extremes, there are many combinations of metals which can be successfully joined by resistance welding as shown in the ac-

Combinations of Metals Suitable for Resistance Welding

METALS	Aluminum	18Cr-8Ni St.	Brass	Copper	Galv Iron	Iron	Lead	Monel	Nickel	Nichrome	Tin Plate	Zinc
Aluminum	▨										▨	▨
18Cr-8Ni St.		▨	▨	▨	▨	▨		▨	▨	▨	▨	
Brass		▨	▨	▨	▨	▨		▨	▨	▨	▨	▨
Copper		▨	▨	▨	▨	▨		▨	▨	▨	▨	▨
Galv Iron		▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	
Iron		▨	▨	▨	▨	▨		▨	▨	▨	▨	
Lead							▨				▨	▨
Monel		▨	▨	▨	▨	▨		▨	▨	▨	▨	
Nickel		▨	▨	▨	▨	▨		▨	▨	▨	▨	
Nichrome		▨	▨	▨	▨	▨		▨	▨	▨	▨	
Tin Plate	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	▨	
Zinc	▨		▨	▨	▨		▨					▨

companying table, reproduced by courtesy of P. R. Mallory & Company.

Although materials having high electrical resistance are preferred for the parts to be welded, other requirements may dictate the use of low resistance materials. The added cost for special tools and equipment for welding materials such as copper, brass, lead and aluminum may make the use of other methods of fastening more desirable. Such factors as the type of weld, thickness of sheets, proximity of other welds and available equipment must be considered.

Flash or Butt Welding

The type of weld governs the cost of the welding operation to a large extent. Flash or butt welding joins the entire cross section and, in general, is restricted to bar stock. The design of the parts for such welding is relatively simple. The parts to be welded are clamped in jaw electrodes as shown in Fig. 3, butted together under a "follow-up" pressure and a heavy current is applied for a short time. This is a low-cost process giving a weld of high strength, but a "flash" of

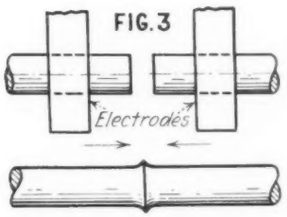


Fig. 3—Flash or butt welding joins the entire cross-section but leaves a ridge at the weld

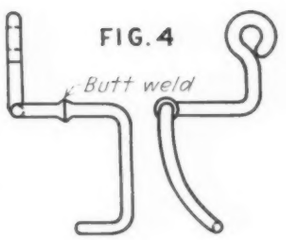


Fig. 4—When made of two parts butt-welded together, the cost of this carburetor lever is much less than when bent from one piece

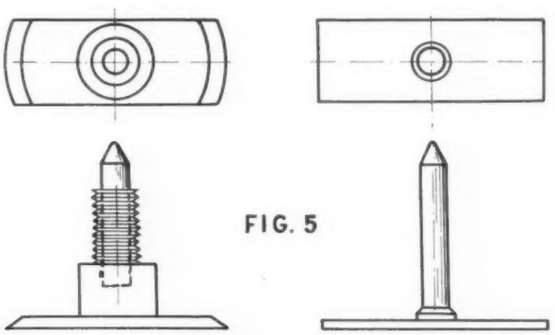


Fig. 5—More than 70 per cent of the weight of the conventional pivot mounting (left) is saved in the butt-welded assembly shown at the right

Fig. 2—Tamper-proof assembly for the glass watt-hour meter cover is provided by welding the brass retaining ring to the steel base

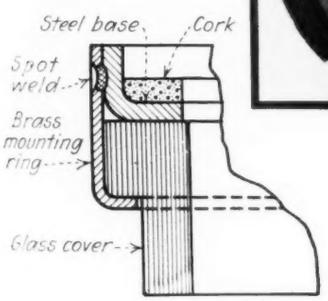


Fig. 7—The projection concentrates the heat, producing a localized weld

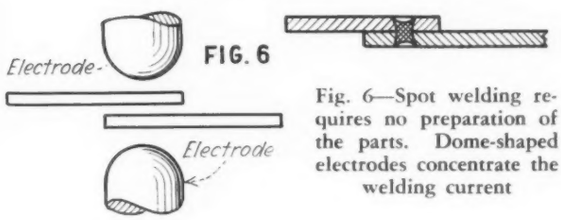
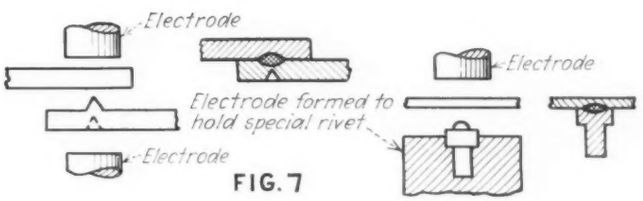


Fig. 6—Spot welding requires no preparation of the parts. Dome-shaped electrodes concentrate the welding current



mushroomed metal is produced which must be trimmed off if it is undesirable. Steel is readily butt welded, and the process is widely used for fabricating rings and chain, and in welding hardened tools to soft shanks. Copper, brass, and many other metals are also butt welded readily. One important application is in joining copper cables to make continuous electrical conductors.

Butt welding can also be used to simplfy the construction of complicated bent forms, reducing the cost of the part. For example, the carburetor part shown in Fig. 4 has formed bends in three planes, calling for an expensive forming operation. By making the part of two pieces to be butt-welded together, the bends can be formed in the separate pieces at low cost. However, lack of accurate control over the amount of "mushrooming" may limit the application of this method.

By butt-welding a jewel-bearing pivot to a strip of non-magnetic alloy, as shown at the right in Fig. 5, the weight of the assembly was reduced to 27 per cent of the former design shown at the left. By reducing the weight carried on the pivots, the welded pivot resulted in less friction and made possible a distinct gain in the performance of the electrical instrument, in addition to reducing the cost of the assembly.

In spot welding, as shown in Fig. 6, the sheets are overlapped and placed between two electrodes, pressure is applied, and current is passed for a short length of time.

In the cost of preparation of the parts, spot welding

is much cheaper than riveted or bolted assemblies, since no drilled or tapped holes are necessary. Additional parts needed for riveted and bolted fastenings are elminated, and the weld is stronger than soldered joints.

To produce a localized weld between the parts, it is necessary to use relatively thin sheets. In applying spot welding to thick sheets there is a tendency for part of the welding current to spread out and pass through the material at some distance from the electrodes, reducing the heat generated where the weld is desired. If the current is then increased to a value which will produce a welding temperature, the weld will have a rough and overheated appearance, the electrodes may fuse or soften and an explosion may occur. For thick sheets other types of welds are used.

Seam welding—a modification of spot welding—produces a continuous weld between thin sheets. Fixed electrodes are replaced by wheel electrodes and the work is fed continuously between the wheels, a steady welding current being supplied at the same time. This method produces strong, leak-proof joints at low cost but the operation requires special equipment.

Resistance Welding for Thick Sheets

Projection, ridge, or button welding is applicable for sheets too thick to be readily spot or seam welded. Concentration of current and pressure is secured by limiting initial contact between the parts to relatively small areas located at the point where welds are desired. In projection welding, Fig. 7, small projections are made in sheet metal parts with a punch from the back side, or in thicker parts the projections may be formed by machining or with special dies.

Projection welding of "blind" fastenings can be used to improve appearance, also. For example, all fastenings are eliminated from the decorated surfaces of the voltmeter cover shown in Fig. 8. This cover is removable, being held in place by snaps of a type much used in clothing. The male halves on the snaps are projection welded to the back of the cover using a flat electrode in contact with the front surface of the cover. The flat surface of this electrode distributes the welding pressure, leaving no sign of fastenings on the exterior.

In permanently attaching a brass stud to a brass terminal strip, as shown in Fig. 9, soldering, brazing, projection welding or riveting could be used, but soldering was not strong mechanically, while brazing was too expensive. Projection welding required a special welding fixture to hold the stud perpendicular to the strip, and for a riveted assembly a special riveting block and punch was necessary. Between riveting and welding there is little choice except that some factor such as additional handling is involved in taking the parts to the welding machine.

For ridge welding, Fig. 10, the ridges can be made in sheet metal by means of dies, or they can be rolled or drawn on bar stock. The pole pieces of the magnet, Fig. 1, are ridge-welded. Since the pole pieces are cut from drawn bar stock the addition of the ridge does not add to the cost of the part. Although the additional operations required to provide these projections or ridges ordinarily will add somewhat to the cost of the welded assembly, the elimination of tapped holes, studs and other parts offers cost advantages favoring ridge welding of bulky parts in many instances.

Button welding, as shown in Fig. 11, is applicable for joining relatively thick parts where projections or ridges cannot be formed readily. Assembly by button welding is somewhat slower and, therefore, more expensive than other welding methods because it is necessary to place the buttons in position before each welding operation. But in fabricating parts from materials that are difficult to machine or in building up

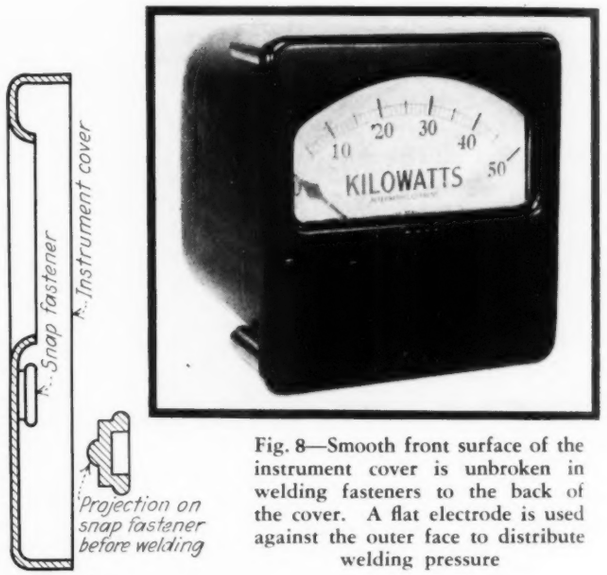


Fig. 8—Smooth front surface of the instrument cover is unbroken in welding fasteners to the back of the cover. A flat electrode is used against the outer face to distribute welding pressure

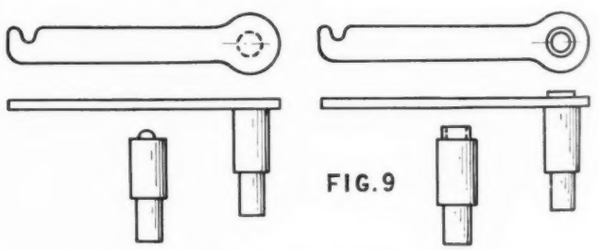


Fig. 9—Assembly of this instrument terminal by projection welding (left) or riveting (right) is equally satisfactory. Other factors, therefore, determine the method of assembly selected

experimental models of new designs, the use of button welds will often reduce machining costs and save time as well.

Welding machines for any of the various types of welds perform two functions. First, the metals are heated to the welding temperature by an electric current supplied for a predetermined time interval, and second, pressure applied at the point to be welded fuses the parts together. Heat required for the weld is generated in the metal itself, the amount of heat depending on the current, the contact resistance between the parts to be welded, the electrical resistivity of the metals, and the length of time the current is supplied.

Adjustable current at low voltage is supplied by taps in the primary or line side of a transformer in the

welding machine. The secondary winding of the transformer has one or more turns of heavy copper wire connected to the welding electrodes with short leads to reduce the electrical resistance of the circuit. The maximum current obtainable at the electrodes is limited by the resistance of the circuit and also by the temperature rise in the transformer, leads and electrodes during the operating cycle, and by the capacity of the power supply line.

The Influence of Welding Pressure

Pressure between the two electrodes is supplied by an adjustable "follow-up" motion, obtained by means of weights, springs, compressed air or through a mechanical linkage. In the welding machine used for the magnet assembly, Fig. 1, a motor-driven crank oscillates a slide carrying the upper electrodes, each revolution of the crank giving one weld. The upper electrode is held in the slide under an adjustable spring pressure, and the height of the lower electrode is adjusted so that the slide has a small amount of overtravel after the electrodes contact the parts to be welded. Thus, when the heat of the electric current softens the metal between the electrodes the spring forces the upper electrode down, fusing the parts together.

Welding pressure also influences the electrical resistance at the point of contact and therefore affects the amount of heat generated by a given current. With a given current flowing, an increase in the pressure generally produces a decrease in temperature because of better electrical contact between the parts. A decrease in pressure while producing a higher temperature may not be sufficient to fuse the parts together.

With a given current and pressure the temperature obtained at the point of contact is controlled by the length of time the current is supplied. In order to concentrate the heat at the weld, heavy currents for short time intervals are used. Thus, the metal at the point of contact reaches the welding temperature without appreciable loss of heat by conduction.

For metals, such as copper and aluminum, having low electrical resistance and high heat conductivity, heavier currents for shorter time intervals are necessary. Also, for metals rapidly oxidized at welding temperatures short time intervals are preferable. Cams on motor-driven welders, auxiliary motor-operated switches, or electronic timers are used for short interval timing, intervals as short as 1/120 sec. being obtained accurately by electronic means.

For most welded designs the exact value of power, pressure and time required are determined experimentally. Tables are available for simple spot welding of thin sheets, but for other jobs experience and experiment will be necessary. The best procedure is to make up from 5 to 100 sets of parts from a preliminary design and develop the proper settings of the welding press.

Welding Machines with Multiple Electrodes

For example, the permanent magnet shown in Fig. 1 is welded on a 60 kva. motor-driven welding press having water-cooled electrodes. The approximate settings used are:

Pressure	600 pounds
Power	40 kva.
Timing	1/6 sec.

In setting up new work, a temporary copper fixture will sometimes be needed for making the experimental welds. After the design is completed the copper fixture can be replaced by an accurate fixture tipped with special electrode alloys at points where the heat and pressure cause excessive wear or softening of the copper. There are available several special low-resistance electrode alloys that will withstand high compressive stresses at high temperature.

Where more than one weld is involved in an assembly, the possibility of using a design incorporating simultaneous welds depends upon the effect of the welds upon each other as well as the shunting effect of the first weld upon successive welds. In general, welds located close together must be made simultaneously while welds well isolated from each other can be made independently, the spacing between simultaneous welds depending on the electrical resistance of the material between the welds. For some designs, such as the welded permanent magnet in Fig. 1, welds can be made in series; in other instances the assembly can be welded on multi-point machines having several sets of electrodes insulated from each other at the transformer.

Resistance welding operations other than spot welding usually will require special welding fixtures and electrodes which for some operations must be water-cooled. If there is not sufficient work to keep these expensive tools in operation, the unit cost of special welds will be prohibitive. However, welding machines are ordinarily adjustable for a wide range of pressure, timing, power and setting of electrodes. This adjustability makes it possible to perform a wide variety of welding operations on a single machine, a factor which tends to reduce the cost of welded assemblies.

FIG. 10

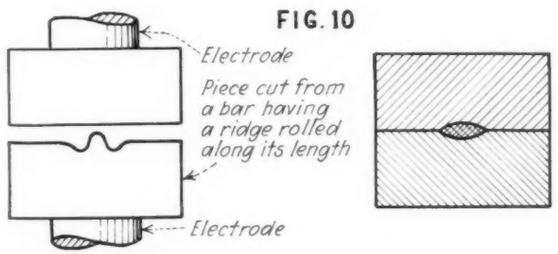


Fig. 10—Machined or extruded ridges can be used instead of a pointed projection to concentrate the current and pressure where the weld is required

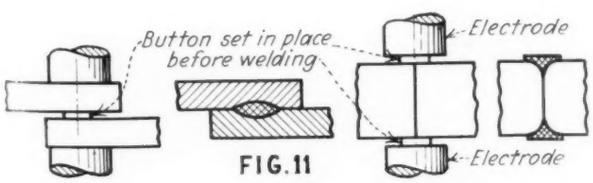


FIG. 11

Fig. 11—Hardened steel parts assembled by button welding are annealed or softened only at the point of fusion

DEVELOPMENTS IN PRODUCT DESIGN

Crankless Diesel Engine

Eliminating crankshaft, connecting rods, cylinder heads, gaskets, valves and camshafts, the new high-speed, crankless, horizontal diesel of the Sterling Engine Company represents a successful development of the wobble-plate principle. This engine weighs but 13 to 20 lb. per hp., depending upon the size and service, as contrasted with the usual 40 to 50 lb. It is a 2-cycle unit with a guaranteed fuel consumption of not more than 0.43 lb. per hp.

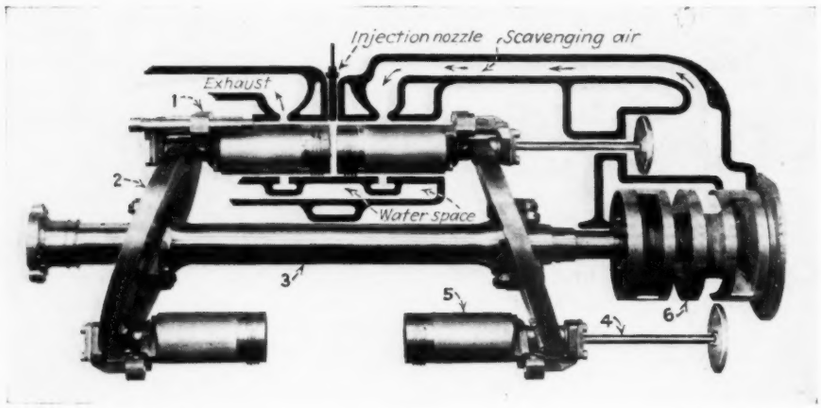
Each of the horizontal cylinders contains two opposed pistons, making the engine equivalent to the usual 2-cycle 8-cylinder engine of corresponding bore and stroke. The cylinders are arranged about a straight drive shaft which carries an inclined disk or wobble plate at each end.

Mechanical bearing units, transferring the power from the pistons to the wobble plates, utilize the same principle as the Kingsbury thrust bearings. A bridge from the piston base passes over the rim of the disk and carries two bearing surfaces, the thrust surface being a slipper of rectangular shape with a babbitt bearing surface mounted on a half sphere. The other bearing, serving only to position the piston, is a bronze base half sphere. This construction provides a universal mounting which permits the bridge to adapt itself to the motion of the inclined disk. Two short rods extending through guide holes in the bridge take all side thrust leaving only axial loads on the piston.

Following the conventional 2-cycle design, the valves are replaced by circumferential ports. Scavenging air is provided by a pressure pump of the piston type, the piston being mounted directly on the end of the bridge. Distributed by a rotary valve at the end of the main drive shaft scavenging air enters at one end of the cylinder and leaves at the other giving uniflow charging and scavenging.

Separate rows of intake and exhaust ports extend completely around the cylinder, and the inclined disks are arranged so that the scavenging port opens and closes after the exhaust port, providing a charge of exhaust-free air for the next explosion. This out-of-phase relation also provides a longer compression interval at approximately constant volume.

The horizontal opposed movement of the pistons is said to provide greater smoothness and less vibration. With no crankshaft, piston side-thrust is eliminated, reducing piston and ring wear and eliminating torsional vibration.

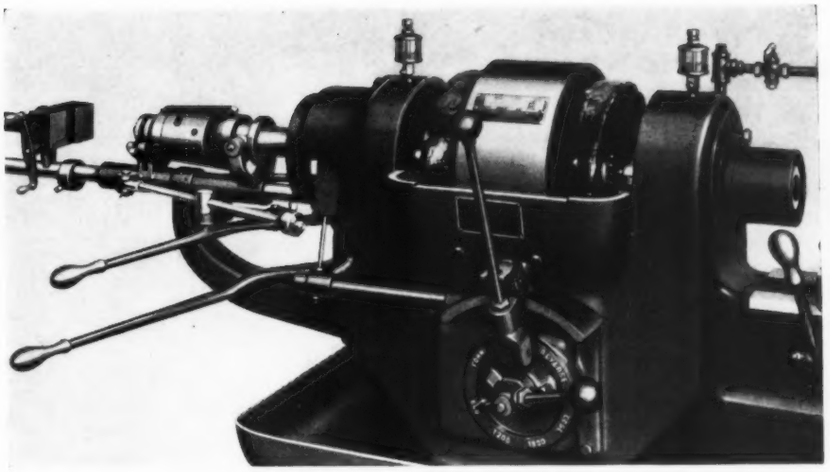


In the Sterling crankless diesel the piston thrust is transferred by the bridge *1*, to the wobble plate *2*, rotating the straight shaft *3*. Scavenging air is supplied by the piston and shaft *4*, direct connected to the power piston *5*, and is distributed to the cylinders by the valve *6*

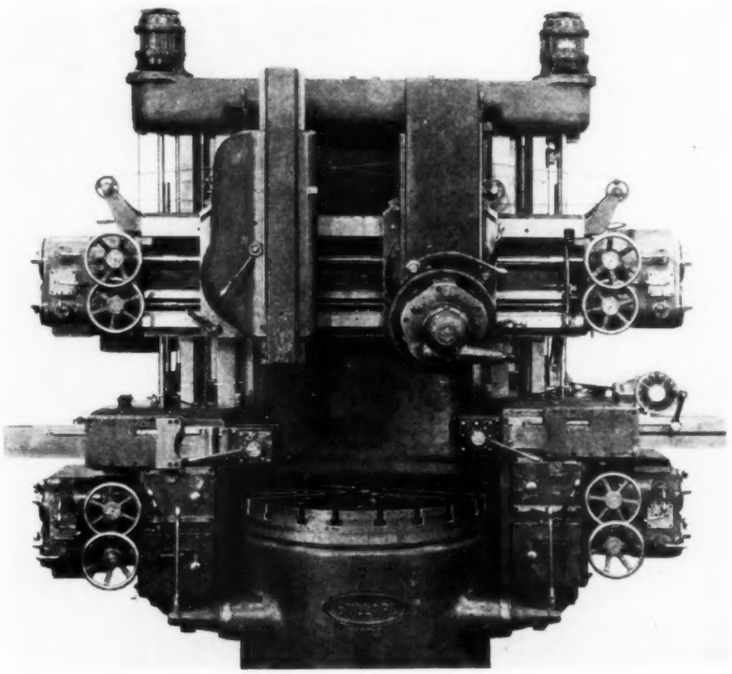
• Ball Bearing Chain Hoist

By carrying all moving parts on precision ball bearings with integral grease seals, the efficiency of the new Wright chain hoist is said to be increased by 10 per cent over former models. To make the hoist weatherproof, oil holes are closed by ball spring covers, and exposed parts are hot-dipped galvanized.

Ball bearings in this hoist increased the efficiency 10 per cent



Mounted on the lathe spindle, the built-in alternating current motor in the Warner & Swasey lathe provides four speeds in either direction



To simplify the operation of the new Bullard lathe, the machine is equipped with a hydraulically operated speed shift mechanism with a visible dial indicator

• Hydraulic Shift for Change Gears

Demands for increased flexibility and ease of operation are met in the new Bullard vertical lathe by a simplified speed changing mechanism operated by hydraulic pressure.

Speed selection in the main drive is by a direct reading dial controlled by a crank lever, the selection being made while the machine is in operation, if necessary. When ready to make the speed change the clutch is disengaged, bringing the table to a full stop. Change gears are then automatically shifted by hydraulic pressure.

Separate feed works with manually selected speeds provide 16 speeds each for the cross-rail and the two side heads. Accessible trip levers on the front of each feed works stop the feed instantly, and an overload device stops the machine if excessive feed or jamming occurs.

Rapid power traverse is provided by separate motors whose controllers have handles designed to disconnect the power when hand pressure is removed. As a further safety measure, limit switches and safety stops are provided, as well as interlocks at each binder station.

Filtered oil under pressure is supplied to the spindle bearing and all bearings in the headstock and side heads. Individual feed works each have their own lubricating system independent of the main supply.

To give the operator full control of the machine from either side, clutch levers are located on either side of the table out of the way when loading or unloading the machine.

• High-Speed Built-In Spindle Motor

To obtain higher spindle speeds with less vibration, a new lathe built by Warner & Swasey incorporates a built-in motor having the rotor mounted on the lathe spindle, and the stator held in the head housing. This motor mounting with its built-in control is shown in the accompanying illustration.

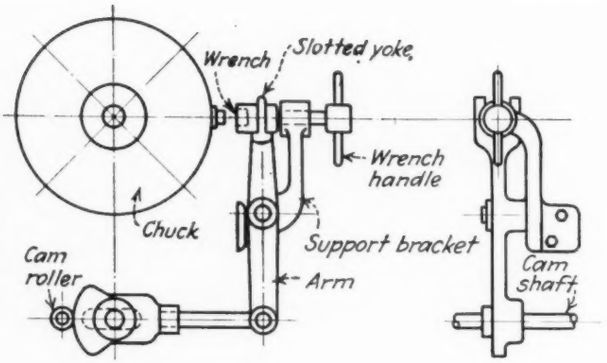
Four speeds in either direction are available, selected by a controller built into the head below the motor. Speeds of 600, 1,200, 1,800 and 3,600 r.p.m. are selected by the short lever, while the longer lever controls the direction of rotation and a band brake.

Power available at each speed varies from 1 hp. at 600 r.p.m. to 3 hp. at 3,600 r.p.m. Cooling air is supplied by a fan mounted on the rear of the spindle, supplying air to the motor winding and the spindle bearings.

Front spindle bearings are preloaded precision ball bearings taking both radial and thrust loads, while straight roller bearings are used at the rear. To eliminate excess oversupply of lubricant, oil-cups automatically feed oil to these bearings only when the bearings are at the running temperature.

• Users of Rust Proofing Increase

Trends toward the adoption of chemical rust-proofing processes have been well maintained throughout the depression years, according to information from the Parker



In addition to its convenience, the cam-operated wrench support is foolproof as the wrench can be brought into engagement with the screw only during the loading period

Rust Proof Company. The total tank capacity installed at the end of 1933 is nearly twice the 1929 capacity. Although the number of licensed customers increased during 1933, there was a slight reduction in tank capacity as a result of improvement in the chemical efficiency of the process, which reduces the processing time from ten minutes to five minutes. These improvements have also reduced the installation costs of the equipment for a given output, cost of steam for heating the solutions and the labor costs.

• Camera from Molded Plastics

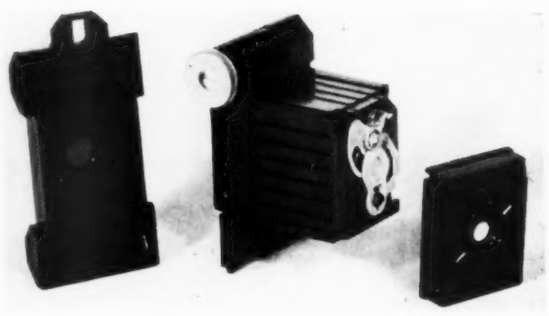
Made possible by recent developments in the molding of synthetic plastics, the small camera shown here, designed by the Universal Camera Corporation, consists of three molded Textolite parts, five stampings and one die-casting.

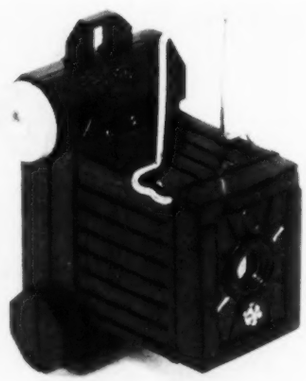
The shutter mechanism made up of three punchings and a spring, is fitted over two molded bosses on the front of the body. On the inside of the front cover a mating boss fits over the extending lower boss holding the shutter assembly in place, and a special screw passing through these bosses holds the front cover. The close tolerances required to obtain the proper shutter action and satisfactory fit of the molded parts presented a difficult problem in die design. But tolerances on the molded parts and punchings are so well controlled that no machining is required, the parts being assembled as received from the dies.

The lens is held in a recess inside the body by a special spring washer clamped by a nut on the end of the screw holding the shutter assembly. Complete front assembly of the camera is thus confined to one operation in which the shutter parts are placed on the bosses, the front cover attached and a single screw inserted and screwed into the nut back of the lens washer.

To prevent light reflections, the inside of the camera is sprayed with a special black lacquer that gives a dull finish, the lens retaining washer is Parkerized, and the nut holding the lens washer is given a blued-steel finish. All flashes, ejector pin bosses and seams are eliminated on the outer surfaces of all three molded parts, giving an excellent finish. The shutter parts, spooling mechanism parts, all springs and the die-cast knob are cadmium plated to withstand a 50-hour salt-spray test.

Two small bosses extend from the inside of the flange on each side of the cover and mate with recesses in the side of the body flange, thereby holding the back cover in place by a snap fit. Also, there is an undercut in the wall of the film window to hold a small disk of red celluloid, which is merely snapped in place. The bosses on the back cover together with the undercut in the wall of the film window cause the back cover to adhere to the mold plunger when it is withdrawn from the mold. Thus, no ejector pins are required against the outside surface of the cover.





This miniature camera consists of but three molded parts and five stampings. Highly developed die design makes it possible to assemble the three molded parts and the shutter mechanism without any machining

• Foolproof Chuck Wrench

In machining metal golf-club heads at one handling, the heads are loaded into jaws at the eighth station on an eight-station automatic indexing machine. Considerable time was lost by the operator during each loading operation in fitting a socket wrench on the chuck screw. By supporting the wrench in a sleeve in position to center over the screw, most of this time was saved. But, when the wrench was not removed before indexing, there was a possibility of damaging both the chuck and the wrench support.

To make the clamping operation foolproof, the wrench is pulled away from the chuck by a cam just before indexing, as shown in the accompanying diagram. This design allows the operator to engage the wrench only during the loading period, in addition to saving time and adding to the convenience of the operation.

• Hydraulic Stoker Drive

A hydraulic transmission as shown in the accompanying drawing is used for driving the coal feed screw of the Schwitzer-Cummins stokers. The main shaft carries the fan, driving pulley and oil

pump and is driven by an electric motor with a *V*-belt drive. The pump draws oil from the reservoir in the bottom of the case and forces it into the hydraulic cylinder. Oil is admitted to and discharged from the cylinders through cylinder ports that are covered and uncovered by the reciprocating trunk piston. The movement of the piston is transmitted to an arm which carries a pawl that actuates a ratchet wheel mounted on the extending shaft of the coal feed screw. The ratchet wheel advances only one tooth per piston stroke.

Turning effort on the feed screw is limited to a predetermined load by the adjustment of a by-pass valve connected to the cylinder. Thus no damage can occur to the mechanism nor can the electric motor be overloaded when an obstruction locks the feed screw. And as soon as the obstruction is removed the transmission resumes operation without any further attention.

Change in rate of coal feed is accomplished by a simple adjustment that varies the clearance volume behind the piston. This changes the frequency of the power stroke because the clearance volume behind the piston must be filled each time by the oil from the constant-volume pump before the power stroke begins. Five rates of coal feed are provided.

On the return stroke of the piston a portion of the oil discharged from the cylinder is diverted to flood all of the moving parts, assuring complete lubrication.

• Compact Three-Stage Speed Reducer

Reversible operation and a torque of 1,200,000 in.-lb. within a restricted space were required for a speed reducer in the control element for an Allis-Chalmers water-wheel generator. Meeting all these requirements, the three-stage Heliocentric speed reducer illustrated here, incorporates a speed reduction of 29,000 to 1 within an outside diameter of 18¾ in. Including the 2-hp. flange-mounted motor, the unit is only 36 in. in length.

Each of the three Heliocentric elements consists of an eccentric on the input shaft operating a series of plungers extending radially outward through a retainer. These plungers engage gear teeth of special contour cut in an internal-gear ring. The number of plungers is one less than the number of internal-gear teeth so that the gear ring advances one tooth for each revolution of the input shaft.

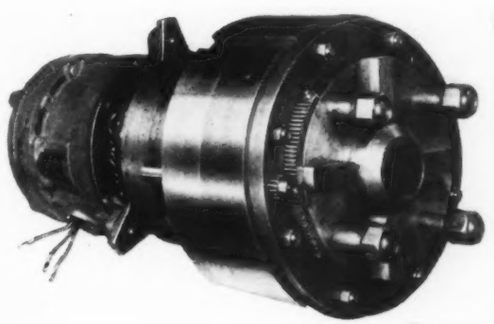
In this three-stage unit, the output shaft of the first stage becomes the input shaft of the second, the same applying for the third element. Since the torque load is distributed at many points around the internal gears, the high torque output is obtained with comparatively low working stresses in the gear teeth.

• Corrosion-Resisting Materials in Dryers and Coolers

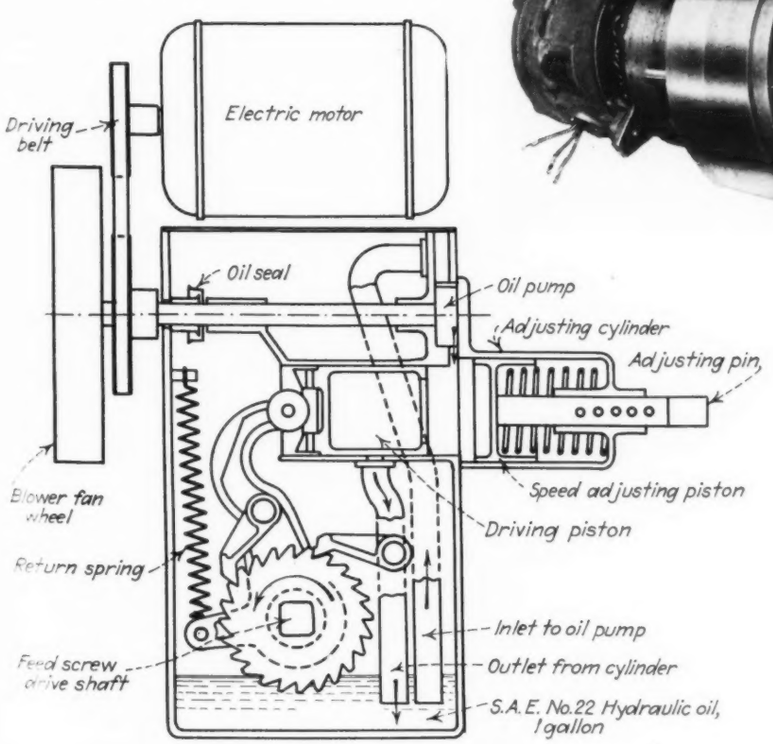
Severe corrosion conditions met in the use of a large dryer built by the C. O. Bartlett & Snow Company dictated the use of corrosion-resisting material. However, the size of the dryer, 35 ft. long and 6 ft. in diameter, made the use of low cost materials necessary.

For the cylindrical walls of the dryer, stainless clad plates, ¼ in. thick were used, the stainless side being 18-8 chrome nickel steel, 0.050 in. thick. All joints were arc welded. End plates were solid 18-8 stainless steel, riveted to angle irons of the same material. Cast-iron sealing rings were covered with stainless-clad plates, also welded in place.

Supplied with the dryer, a cooler 5 ft. in diameter and 30 ft. long was built of heat-treated aluminum alloy 51ST plates, ¼ in. thick, fabricated entirely by riveting. The heat-treated alloy was corrision resisting and sufficiently hard to resist abrasion and, in addition, the high heat conductivity of the alloy made possible a rapid cooling rate.



Within a restricted space a reduction ratio of 29,000 to 1 is provided by this three-stage Heliocentric speed reducer built by the Universal Gear Corporation



Hydraulic transmission of the Schwitzer-Cummins stoker allows five changes in speed, has overload protection and flood lubrication

Avoiding Stress Concentration By Using Less Material

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Westinghouse Electric & Manufacturing Company

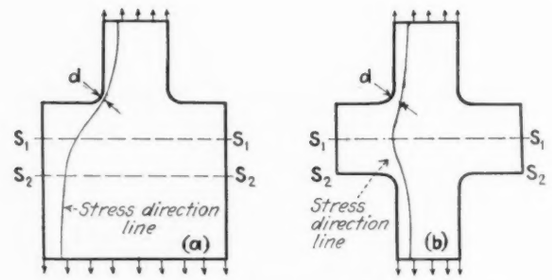


Fig. 1—Stress concentration is proportional to the abruptness with which the stress lines must bend, or inversely proportional to *d*

AN INCREASE in load is generally taken care of by an increase in the proportions of the part. But this is not always true. For instance, it is well known that when inertia forces are acting on a machine part, an increase in dimensions does not necessarily decrease the unit stresses. Thus, doubling the thickness of a rotating disk does not increase the strength of the disk against bursting from centrifugal force. The only solution to such a problem is to employ material of a higher strength to weight ratio, or change the design completely.

Paradoxical as it might seem, it is sometimes necessary to cut away material in order to get a stronger part. An elementary example of this is shown in Fig. 1. For a given load, the maximum stress in the part shown at *b*, which has less material, is less than that shown at *a*, a fact that has been proved photoelastically, and which can be explained readily.

The law of continuity, a fundamental principal in hydrodynamics, may be applied also in elasticity as a first approximation. This law of continuity states that the velocity of fluid flow is inversely proportional to the distance of the stream lines from the boundary.

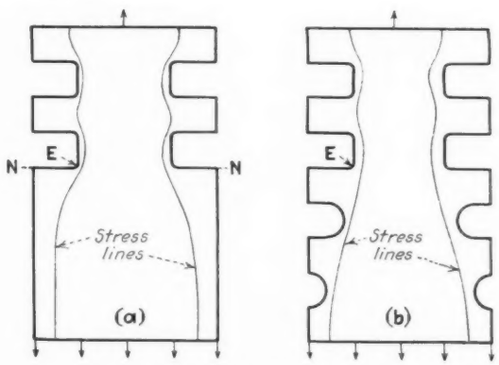


Fig. 2—With additional notches or grooves as shown at *b*, the effect as to stress concentration is similar to that of a taper

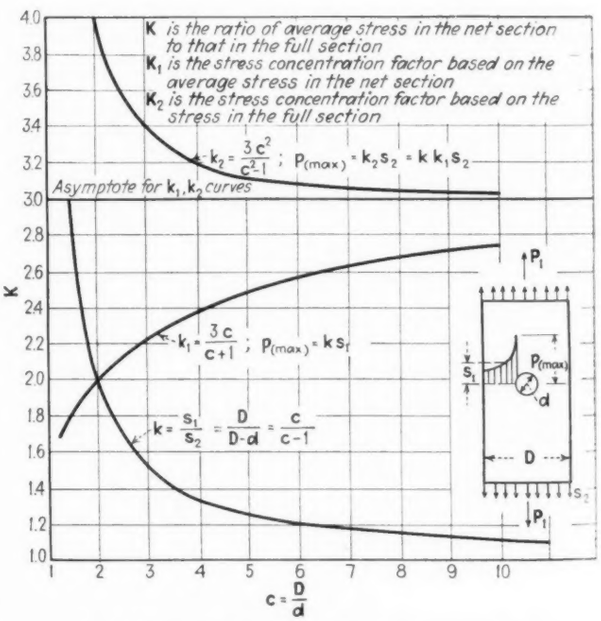

K is the ratio of average stress in the net section to that in the full section
K1 is the stress concentration factor based on the average stress in the net section
K2 is the stress concentration factor based on the stress in the full section
k2 = 3c²/(c²-1) ; P(max) = k2 s2 = k k1 s2
Asymptote for k1, k2 curves
k1 = 3c/(c+1) ; P(max) = k s1
k = s1/s2 = D/(D-d) = c/(c-1)
K
c = D/d
P1
P(max)
d
D
s1
s2

Fig. 4—Curves giving the stress concentration factor for holes of diameter *d* in plates of width *D*

Fig. 3—Stress concentration at the bottom of keyways in hubs can be relieved by adding grooves as shown

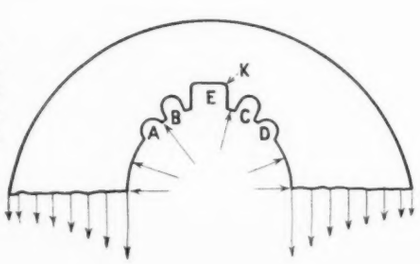


Fig. 5—If the center distance between holes is equal to or less than twice their diameter, the maximum stress will be decreased 20 per cent or more

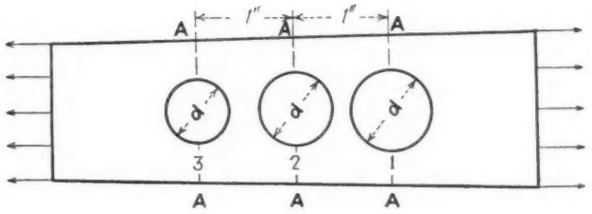
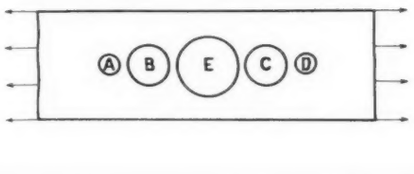
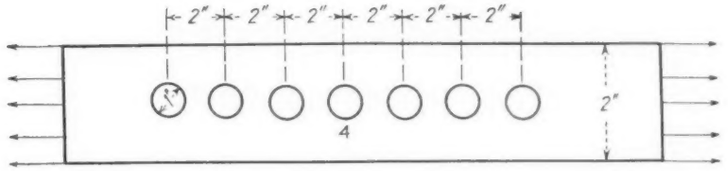


Fig. 6—If the distance between centers is equal to or more than twice the diameter of the hole, the maximum stress will not be affected

Fig. 7—Although only one hole may be required, by adding a series of holes of decreasing diameter, the stress concentration will be minimized





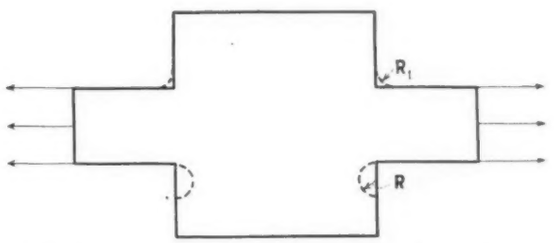


Fig. 8—If external fillets cannot be used to reduce the stress concentration, the same results can be obtained by using undercuts

TABLE I

Dimensions of specimens shown in Figs. 5 and 6, factors of stress concentration k_1 calculated from Equation (2) for a single hole and k_M obtained from photoelastic tests.

Fig.	Point	$D = A - A$	$c = D/d$	k_1	k_M	k_M/k_1
5	1	1.618 in.	2.00	2.00	1.60	0.80
5	2	1.563 in.	2.09	2.03	1.50	0.74
5	3	1.490 in.	2.38	2.11	1.80	0.85
6	4	2.000 in.	2.00	2.00	2.00	1.00

TABLE II

Reduction of area of test specimens shown in Fig. 9

Test Piece	Original Dimensions and Area			Area of Fracture
	Width in.	Thickness in.	Area sq. in.	sq. in.
1	0.501	0.125	0.0627	0.0267
2	0.502	0.125	0.0628	0.0373
3	0.500	0.126	0.0630	0.0395
4	0.500	0.125	0.0625	0.0417
5	0.500	0.125	0.0625	0.0417

In elasticity this is analogous to stating that the stress is inversely proportional to the distance of the stress direction line from the surface. Thus, a stress direction line, as in Fig. 1*a*, shows a greater bow at the section $S_1 - S_1$ than is shown by the corresponding stress direction line in Fig. 1*b*. It is, therefore, reasonable to assume that for corresponding stress direction lines the distance d from the stress direction line to the fillet contour is correspondingly smaller in Fig. 1*a* than in Fig. 1*b*. Consequently the maximum fillet stress is larger in Fig. 1*a* than it is in the construction shown in Fig. 1*b*. Thus, by the removal of material below the section $S_2 - S_2$, a "disburdening" or a diminishing of the concentrated stresses in the fillet is achieved. This same principle can be applied in a great number of designs, some of which will be described.

Expanding upon the principle, it can be generalized that if it is necessary that a section be weakened by a notch or a fillet the stress distribution can be improved by taking away additional material in neighboring sections. This is illustrated in Fig. 2. At *a* is shown a plate that has been notched, or a round bar that has had a circumferential groove cut into it. Referring to the section $N - N$ in Fig. 2*a*, for a given tension the stress concentration at the corner E is greatly reduced by the addition of the two pairs of notches or grooves as shown in Fig. 2*b*. It should be noticed that the bottoms of the grooves in Fig. 2*b* are rounded. It will be observed that in their effect upon the stress line, the successive fillets of decreasing depth are equivalent to a taper.

Frequently it is necessary to have a keyway in hubs that are made a shrink fit on their axles. The shrink fit stresses will cause stress concentration at the root at the keyway. This may be greatly relieved by providing additional notches as shown in Fig. 3, where E is the keyway and notches A, B, C, and D serve to relieve the stress concentration at K.

Holes are another type of "stress raisers" that may seriously impair the strength of a structure. Not only do they decrease the cross-sectional area, but as in fillets, the narrowing of the stress direction line creates a concentration of stresses such as indicated in Fig. 4, wherein is shown the stress distribution across the section. The maximum stress can be computed by using the factors as obtained from the curves in Fig. 4.

For a single hole, the computed average stress, based on the net cross-sectional area is:

$$s_1 = \frac{P}{(D - d)t} \qquad (1)$$

In the above equation P is the load, D is the width of the member, t is its thickness and d is the diameter of the hole. The true maximum stress is obtained by

multiplying the quantity s_1 as obtained from the above equation by the factor of stress concentration k_1. For a single hole the equation for the factor of stress concentration is:

$$k_1 = 3c / (c + 1) \qquad (2)$$

In this equation c is the ratio of the width D of the member to the diameter d of the hole, as shown in Fig. 4. The equation was derived and checked photoelastically by Professor Coker and his co-workers of the University College in London.

In order to study the effect of a series of holes, or "holes in series," photoelastic measurements have been made by the writer using models such as shown in Figs. 5 and 6. The results of such tests are given in Table I. From these results it can be concluded that if the distances between centers of the hole are twice the diameter of the hole, or more, the maximum stress is about the same as for a single hole. If, however, this distance is less, as in Fig. 5, the maximum stress at the center hole may be decreased by as much as 20 per cent, this figure even allowing 6 per cent for possible errors. The first and last hole also show a decrease in the maximum stress of about 10 per cent. This suggests that although conditions really require only one hole, additional holes of smaller diameter should be provided as shown in Fig. 7, thereby reducing the stress concentration.

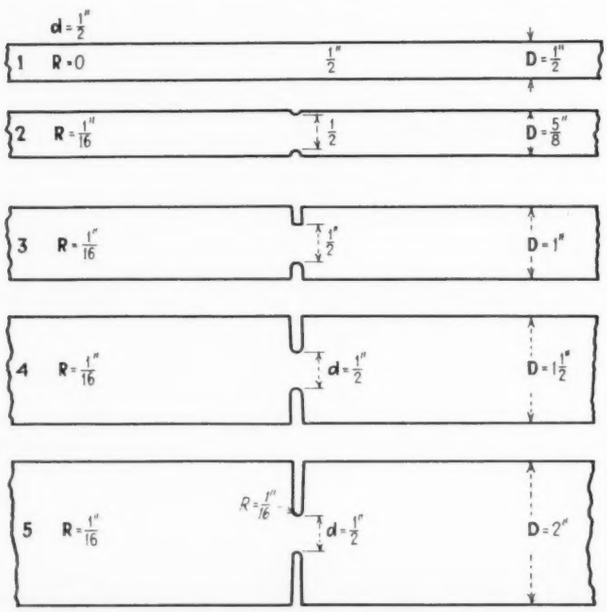
Another example wherein the strength of the member is increased by removing material is shown in Fig. 8. In the sharp corners of such a design, the stress direction lines pass directly through the corners so that theoretically infinitely large stresses must be expected even for small loads. But under some conditions it is not possible to provide an external fillet. In such cases an inside fillet or reverse fillet can be provided to relieve excessive stress concentration.

If the material of construction is brittle, the strength of the member will depend directly on the maximum stress. Similarly, if the load is dynamic, the maximum stress will govern the load-carrying capacity of the member. It is under such conditions that the equations given should be applied.

But if the material is ductile and the fluctuation of the load is relatively small, it is not advisable when designing the member to be guided solely by the maximum stress at certain points. Under such conditions the value of the procedure as outlined becomes questionable. At the same time, it would not be wise to completely ignore stress concentration even with ductile material and static load, because whenever stress-raisers are present the structure as a whole behaves as if the material were brittle. This has been demonstrated by experiment.

Stress Raisers in Ductile Material

To demonstrate the effect of stress-raisers on ductile materials, five test pieces, as shown in Fig. 9, were made of a ductile material. These test pieces were subjected to tensile load in an Amsler testing machine. The stress-strain diagrams obtained are shown to the right of the figure and the reductions in net area are given in Table II. As shown by the curves, the proportional limit for test pieces No. 3 to No. 5 is greater, while the ductility decreases. Test specimens 3, 4 and 5 broke in an abrupt manner characteristic of a brittle material. From this it can be seen that a machine member without spots of stress concentration, will show indications of subsequent failure by excessive deformation, whereas a similar part with stress-raisers will break suddenly without warning. By making the design changes as outlined in this article, it may be possible to change the stress-strain diagram for specimens 3, 4 and 5, to a diagram similar to that given by specimen 1 or 2. Whether such changes are warranted in the case of a ductile material under static load will depend on other considerations affecting the problem.



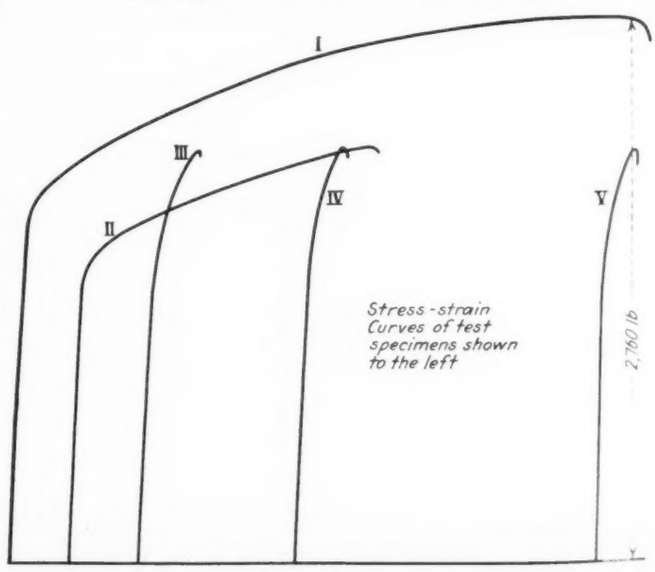


Fig. 9—These five test specimens were used to determine the effect of stress concentrations in ductile material under a steady load. The corresponding stress-strain curves are shown to the right

Engineering Data for Designing Molded Plastics—II

F. W. WARNER
Engineer, Plastics Department, General Electric Company

Plastic molded knobs, buttons, handles and indicators offer economy, artistic design and wide color selection. Since metal inserts can be incorporated in the main piece, assembly operations are simplified

THE use of properly designed inserts for screw threads, studs, supports, ornaments and similar details incorporated to obtain economy of design or subsequent ease of assembly, or to secure rigidity and compactness of construction, decorative effects and similar purposes, greatly increases the utility and economy of molded parts.

In the design of inserts, care should be taken to provide sufficient anchorage for parts that will be subjected to stresses in service. Inserts are normally anchored by knurls, grooves, projections or similar means. Ample provision must be made for holding the inserts in place during the molding operation. Usual methods are to use plain or threaded pins, recesses in the mold, or special clamping arrangements.

To reduce cleaning costs, inserts should be sealed against compound leakage. This is accomplished by the use of closed-end inserts and making them a snug fit with the mold. A snug fit is best obtained by having the face of the insert fit into a slight recess in the mold surface, which necessitates the stopping of a knurl or hex a short distance below the surface. Studs should have a base of larger diameter than the projecting portion, thus providing a shoulder to rest against the mold.

Location and Support of Inserts

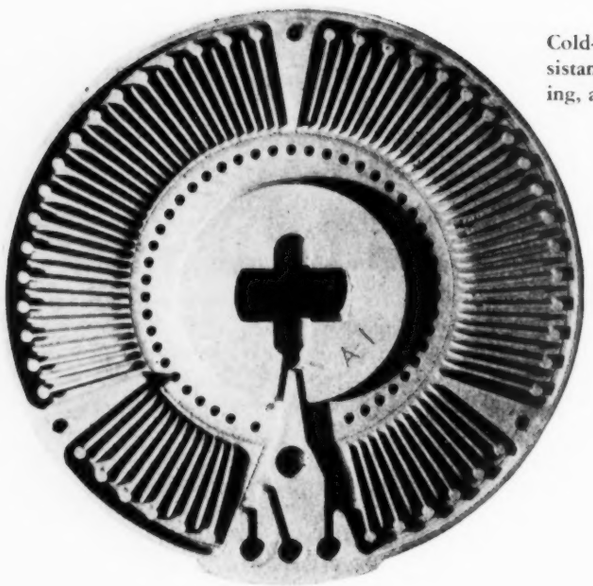
Horizontal inserts require special mold construction and molding procedure because side wedges or pull pins are required. If the inserts are long, additional vertical support must be provided. As the use of supporting pins exposes part of the insert, care must be taken in electrical applications where there is a danger of flashover to provide sufficient clearance between the exposed insert and any "live" parts mounted on the molded piece. The design must also provide a long creepage path to prevent current leakage.

In the design of molded plastic parts the engineer should avoid large masses of metal inclosed by the molded material. Projecting inserts should not be located near edges or corners, and thin layers of compound over inserts should be avoided. Also, the inserts should not be designed with sharp corners. These precautions are necessary in order to avoid reducing the strength of the molding compound. Inserts passing completely through a part and flush on one or both surfaces of the part, should be avoided because the inserts are likely to be crushed during the molding operation. Delicate inserts are difficult to mold because of their tendency to collapse under the molding pressure.

Tapers, Indentations and Projections

Depending on the shape of the molded part, draft or taper may be necessary to permit removal from the mold. A taper of at least 0.005 in. per in. on each side surface of projections or indentations is desirable. Where little or no taper is provided, the removal of the plunger from the part and the ejection of the part from the mold occasionally tends to distort or elongate vertical sections. This consequently introduces variations in dimensions which would not otherwise occur, necessitates increased care in inspection and results in increased scrap losses and higher cost of parts.

Wedges are removable mold parts sometimes required to form projections or indentations, or to hold inserts. They are placed on the sides of the molded parts. Their use complicates mold constructions and results in increased labor for molding, and because of the additional flash lines formed at the wedge joints, increases the cost of cleaning the parts. Such varia-



Cold-molded refractory offers excellent resistance to high temperatures and heavy arcing, as exemplified by this plate-rheostat form

bles as slight displacement of wedges, wedge wear and similar factors are introduced, which necessitates increased tolerances for variations in dimensions. In the interest of low costs and accurate dimensions, parts should be so designed, if possible, to make the use of wedges unnecessary.

It is not possible to make a general statement as to the use of fillets versus sharp corners. From the standpoint of increased strength in both the mold and the molded parts, a fillet is always desirable because it eliminates the concentrated stresses that occur at sharp corners. Also, it is occasionally necessary to have a fillet in order to permit satisfactory flow of compound, as for example, around ribs, bosses and similar places. From the standpoint of mold costs, in some designs a saving can be effected by the use of sharp corners while in other designs a saving can be effected by the use of radii. If the location of the corner is such that the radii must be formed by hand operation instead of machining, the labor cost of the mold construction will be increased greatly.

Where the design has a number of bosses used for mounting the molded parts, it is usually necessary that the surfaces of these bosses be ground flat and to a uniform height. To facilitate this operation, the bosses should project at least $\frac{1}{64}$ in. beyond all other projections. Three mounting bosses are preferable, and with such a design the finishing operation is usually unnecessary.

In the manufacture of molded parts in which heavy sections are jointed to thin sections, the proper curing of the heavy section tends to overcure the thin section. Therefore, where possible, it is often advantageous to insert holes in the heavy section in order to reduce the thickness. This assists in obtaining the proper cure of both the heavy and the thin section by shortening the time required for curing the heavy section. Thus, a more uniform compound strength is obtained and the speed of production is increased.

Where knurling, threading or grooving is needed, care should be taken to provide surfaces that can be molded readily in simple molds. For example, straight rather than diamond knurling should be employed. Diamond knurling requires two or three extra mold parts to allow the mold to be taken apart without marring the knurling. Threads and grooves should be made coarse, rather than small or fine. Where the parts are to be unscrewed from a mold pin, some provision such as a knurl or square side should be made for the unscrewing fixture.

Standard pitch threads are used in mold members, but shrinkage of the molded parts results in threads having less than standard pitch. Therefore, long molded threads will not fit standard threads, but will fit other molded threads. Inserts should be used for threaded holes less than $\frac{1}{8}$ in. diameter because of the difficulty of molding small threads. As a re-threading operation usually ruins molded threads, when a long molded thread must fit a standard thread it is desirable to mold a smooth hole, and tap the hole after molding.

In making the molds, the part number should be cut upon a suitable face so that the number will appear upon the molded piece in raised figures. Depressed lettering or engraving entails increased cost in construction of molds. The location of the part number should be specified on the drawing in order to insure that the location will be such that the number will not interfere with the assembly or operation of a part. Where it is desired to have the top and bottom surfaces perfectly smooth, the letters may be brought flush with the surface by putting them within a recess made for the purpose.

There necessarily must be exceptions to this general procedure of marking. Some parts may be too small to permit them to be numbered properly. Or the design may be of such a nature that no suitable surface is available upon which to place the number. These exceptions should be indicated by the designer with a suitable notation on the drawing.

Types of Cold-Molded Compounds

Cold-molded compounds are divided into two general types: non-refractory, being those compounds containing some organic ingredients; and refractory, being those compounds containing no organic ingredients.

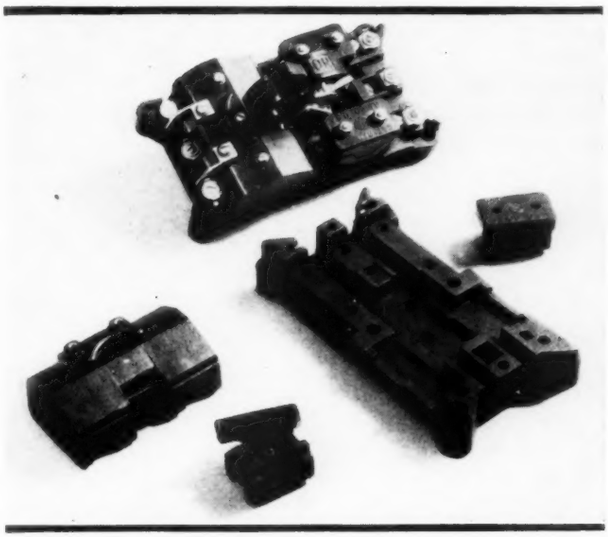
Cold-molded compounds are best suited for the production of parts requiring high heat or arc resistance. They offer the advantages of lower part cost and reduced mold investment where small yearly requirements make multi-cavity molds impractical because of the mold cost. But cold-molded compounds are not suitable for parts requiring high dielectric strength nor where thin sections are required. Metal inserts can be incorporated in the parts to provide simplified and improved designs or simplified assembly processes.

The process of manufacture of cold-molded parts consists of loading the proper amount of molding compound, which is in powdered or granular form, into the cold mold. The mold is then closed to form the part, after which the part is removed in the uncured

condition. The curing of the part is done by a separate heat treatment. Molds containing more than two cavities are usually not practicable in cold-molding, as the flow of the molding compound results in uneven pressure distribution on the cavities.

The general utility non-refractory material is the compound composed of asbestos filler, asphalt and drying oil binder. It can be molded in moderately intricate shapes. It has a smooth dull surface and is available only in black or brown.

The mechanical and electrical properties of cold-molded plastics are not so good as those of the general utility hot-molded material. The cold-molded material has moderate arc resistance, being more arc resisting than hot-molded materials. It resists water, mineral oils and weak acids, but is attacked by alkalies and strong acids. Continued immersion in water has no effect upon the compound. It is not affected by temperatures up to 250 deg. C., or 480 deg. F. The physical properties of this material are also given in Table I (April *P.E.*, p. 139). Typical applications are wiring device parts such as large and small switch bases, conduit outlets, weatherproof sockets, polarity receptacles, switch covers, adapters, contact supports and heater connectors.



Main parts of this switch consist of six pieces of cold-molded material selected for its high dielectric strength

A non-refractory material, having an asbestos filler, asphalt and special resin binder, differs from the asphalt-drying oil type principally in that it is mechanically stronger and is more arc resisting than the asphalt-drying oil type of compound. But this material is not as arc resisting as the refractory type of compound. For physical properties of this material see Table I (April, *P.E.*, p. 139).

Typical applications are bases for industrial control apparatus, miscellaneous switch parts, and applications for which better arc resistance than is possessed by the asphalt-drying oil type of compound is required.

Refractory material having an asbestos filler and inorganic binder has good mechanical strength and good arc resistance. It can be molded in moderately intricate shapes. The surface is smooth and stonelike and gray in color. It is resistant to water, oils, most solvents and alkalies, but is attacked by acids. It absorbs from ten to fifteen per cent of water, which gives it a low dielectric strength. It is not affected by temperatures up to 700 deg, C., or 1,300 deg. F. The physical properties of this material are also given in Table I (April, *P.E.*, p. 139). Typical applications are arc deflectors, rheostat bases, and any applications where heat and arc resistance are of primary importance and where high dielectric strength and low water absorption are not required.

Refractory material has a moderate dielectric strength when dry, hence it is frequently baked and then impregnated with a treating compound or covered with a varnish to exclude moisture. As the treating compound or varnish ordinarily used is not very arc resisting or resistant to temperatures of above 250 deg. C. or 480 deg. F., if the resistance to arcing or high temperatures is not so important as dielectric strength a non-refractory material will often be more satisfactory than a treated refractory material.

Commercial Tolerances

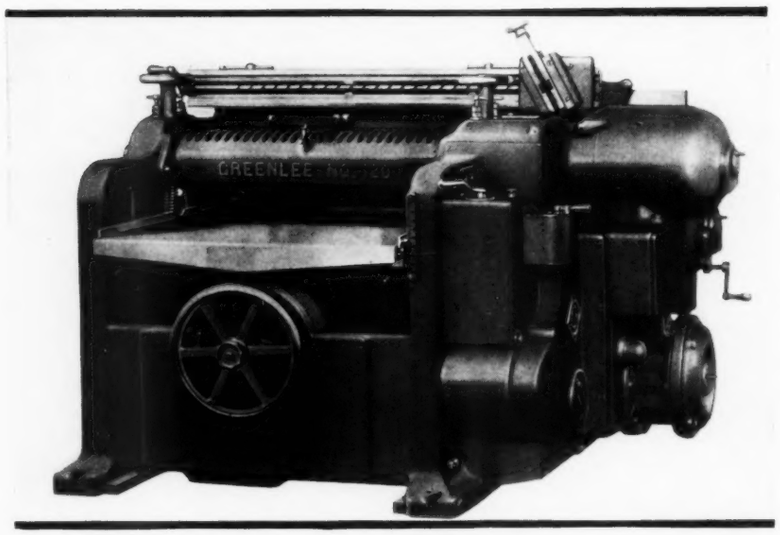
The information given previously on the design of hot-molded compounds is generally applicable to the cold-molded compounds with some exceptions. During the hardening or curing of cold-molded materials subsequent to the molding, there is likely to be variable shrinkage. This, together with increased mold wear, because of the more abrasive character of the compound, results in greater variations in dimensions with the cold-molded than with hot-molded materials. In the accompanying Table IV are given the tolerances which will generally give satisfactory molded parts at a minimum cost. The unavoidable conditions of mold wear and shrinkage in curing preclude attaining closer limits through more accurate mold construction. "Build-up" tolerances on cold-molded parts are plus and minus 0.015 in.

Holes near the edge of a molded part should generally have a minimum side wall thickness of not less than the diameter of the hole. For long cross holes, pins supported on both ends can have a maximum length of one and one-half times the pin diameter. A minimum taper of $\frac{1}{32}$ in. per in. on each side of a projection or indentation should be provided for cold-molded compounds.

TABLE IV
Commercial Tolerances for Cold-Molded Plastic Parts at Minimum Cost

Nominal Dimensions (In in.)	Tolerances above and below Nominal Dimensions (In in.)	Nominal Dimensions (In in.)	Tolerances above and below Nominal Dimensions (In in.)
$\frac{1}{4}$	0.009	5	0.023
$\frac{1}{2}$	0.011	6	0.028
1	0.014	7	0.032
2	0.016	8	0.036
3	0.018	9	0.040
4	0.020	10	0.048

Motor Drives and Controls for Modern Woodworking Machinery



In this double planer, the 4-speed feed motor is controlled by the large drum controller. The small drum switch is for the motor that raises and lowers the bed. Above the large controller are the push buttons for the head motors mounted on the cutter head spindles

BECAUSE woodworking machinery must run at high speeds and without vibration it is perhaps natural that the designing engineers in this field have been most progressive in the adoption of electric motors and controls. In addition to this the combining of operations often requires cutters mounted in various relative positions and in different planes. Direct motor-driven spindles eliminated the many problems of high-speed mechanical transmissions, such as inertia effects, vibrations and the necessity for a great number of bearings.

In the eight-roll electric double planer manufactured by Greenlee Bros. & Company, the top and bottom sides of the stock are planed at various feed rates as controlled by a 4-speed feed motor with drum controller, the feed being also reversible to back out the stock. A feature of the built-in drum controller, as shown in the illustration, is that the case can be swung down so that three sides of the controller mechanism are open for wiring, making adjustments or cleaning.

Mounted on the side of the feed drum controller can be seen a small drum switch which operates a small motor directly geared to the feed screws for raising and lowering the bed of the machine for correct positioning for the thickness of the stock being handled. When the control handle is released it returns automatically to the neutral position. The scale on the edge of the right-hand frame indicates the table setting. The hoist motor is designed so that it will stall without injury to itself or any parts of the machine if the bed is moved too far in either direction or if the operator attempts to raise the bed while the material is in the machine.

The handwheel at the front of the machine can be used for the final precise setting of the bed, giving a movement of $\frac{1}{16}$ in. for each turn. Lengthwise movement of the handwheel shaft disengages the gear to the right hand screw, permitting the left-hand screw to be turned independently for levelling the bed to 0.00075 in. accuracy.

Head Motors Drive Spindles Directly

The pinion of the 4-speed feed motor engages a gear that is connected to a shaft by a safety slip clutch. A train of three gears mounted on an arm that can pivot on the shaft serves as an "expansion linkage" to transmit the drive to the top feed rolls and maintains complete meshing of the gear teeth.

Directly above the feed motor drum control lever is shown the push button control station for the head motors that drive the cutter head spindles directly. Depending on the size of the machine, a 15-, 20- or 25-hp. motor is used for the top head and a 10-, 15- or 20-hp. motor for the bottom head. Their normal speed is 3,600 r.p.m., which is appropriate for most conditions. When desired, high frequency motors are used.

The magnetic switches for the spindle head motors are mounted in a cabinet in the rear and the wiring is in conduits that pass through cored channels in the frame.

Anti-friction bearings are used throughout the machine. All feed rolls run in Timken double bearings mounted in vertically adjustable grease-tight housings. Top and bottom heads run in deep-groove precision bearings. All idler gears in the feed gear train are mounted on roller bearings carried on hardened studs supported on both ends.

In the Greenlee electric double-end tenoner, which is built in lengths for handling material from 4 in. to 16 ft. between shoulders, 14 built-in motors, some of which may be for high frequency, are used. The accompanying illustrations show a right-hand front view of the machine, a close-up of the working end of the top and bottom tenoning and scoring saw units, which are accessible by opening up the swinging top pressure beam, and a view of the control cabinet with the covers raised. This cabinet is at the left end of the machine base.

One of the outstanding improvements in this machine is the placing of the stationary column at the right-hand end of the machine. This permitted the grouping of all the push button stations and all the mechanical controls on the right-side of the machine at the regular position of the operator. It will be noticed that all the push button stations are mounted into a specially designed panel.

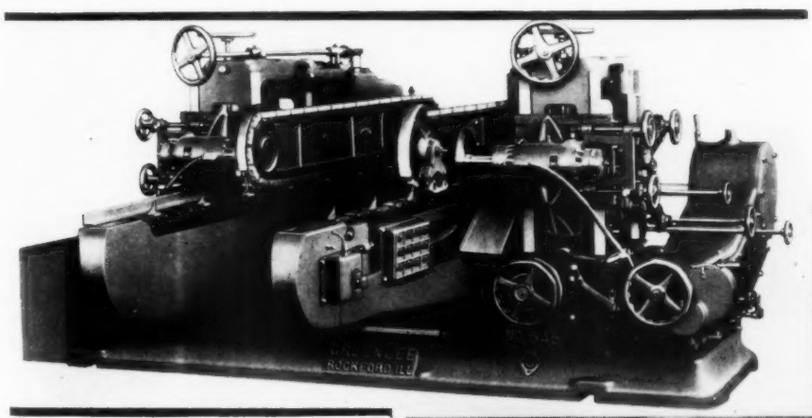
The feed motor mounted in the base of the machine also is used to drive the lead screw which moves the adjustable housing horizontally. The safety type feed clutch is thrown out against spring pressure and rattles in case of overload from stock jamming.

Controls Electrically Interlocked

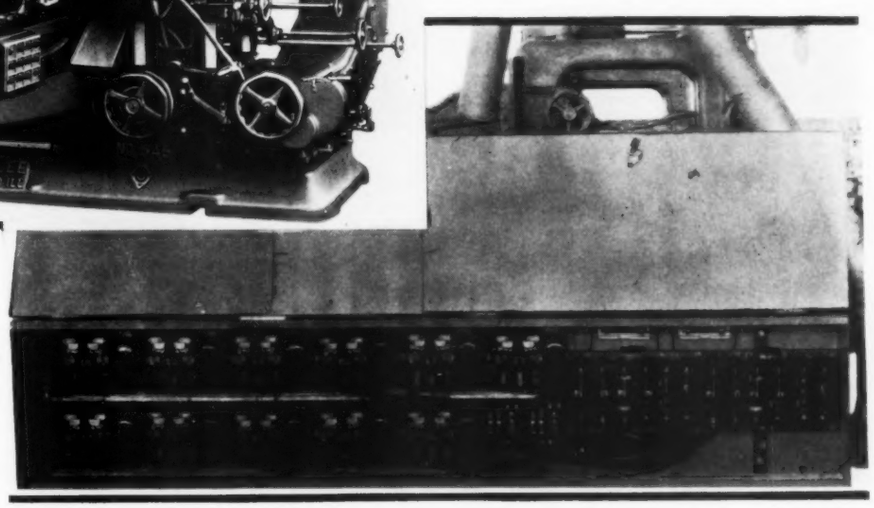
Besides start and stop push button control for each motor, a master push button for stopping all the motors simultaneously is provided. Through electrical interlocks, if one motor cuts out, all the others are stopped.

As in the first machine described, all bearings for spindles and drive shafts are either ball or roller type and are lubricated through high-pressure grease gun fittings, except those in the feed box which are flood lubricated by a gear pump. The oil is pumped from the bottom of the case and is delivered to the gears and bearings through a sight indicator.

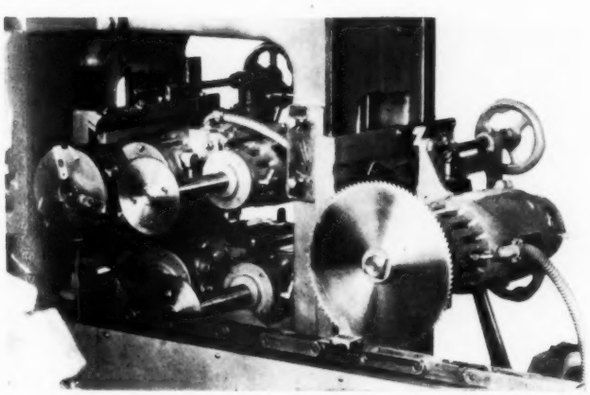
Mechanical oilers are used for the feed chains and tracks and sight-feed oilers for the pressure beam chains which are fitted with rubber hold-down pads to prevent marring the stock. The feed chains run over a pair of 4-toothed sprockets mounted on a $2\frac{3}{4}$ in. shaft carried in two roller bearings in the chain beams and a ball bearing in the feed box which incloses the change-gear transmission. The transmission provides three rates of feed, controlled by a gear-shift handle convenient to the operator.



The fourteen built-in motors of this double-end tenoning machine are controlled from the built-in push button station on the right side of the machine



The magnetic switches for the double-end tenoning machine are inclosed in a steel cabinet mounted on the left-hand end of the machine. The cabinet is shown here with the cover raised



A close-up view of the top and bottom tenoning units and the scoring saw with the top pressure beam swung out of the way

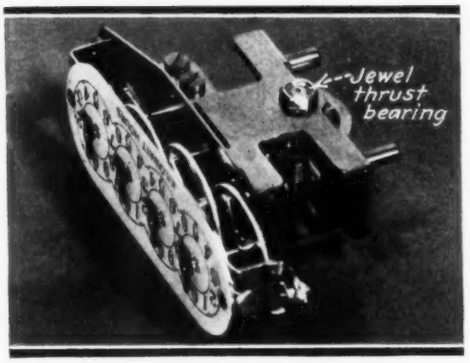


Fig. 1—To reduce friction, a sapphire "end-stone" is used for the worm thrust bearing in this watt-hour meter register

Jewel Bearings For Minimum Friction

THE two types of jewel bearings for thrust loads differ widely in their radial load capacity. The combination hole and "end-stone" jewel bearings (described in the article "Types of Jewel Bearings and Their Mountings," appearing in the March number) can carry appreciable radial loads. A bearing of this type is shown in Fig. 2. It carries the worm thrust load of the shaft in the watt-hour meter register shown in Fig. 1. In this application it is necessary to use a radial bearing in order to maintain proper engagement of the gearing.

But where the smallest possible frictional drag is the first requirement, the V-jewel is preferable, although this bearing has relatively large end play and radial clearance. The V-jewel bearing, shown in Fig. 3, is substantially a flat end-stone provided with a conical depression to hold the pivot in the bearing. Applications of bearings of this type are limited by the fact that end play and side clearance each vary with the magnitude of the other.

In V-jewel bearings designed for minimum friction, a balance must be obtained between two conflicting requirements. Reduced friction is secured by decreasing the area of contact between pivot and jewel. But the unit pressure under a given load also increases with reduced contact area. If the elastic limit of the pivot material is exceeded permanent deformation occurs resulting in greatly increased friction. A pivot bearing designed for minimum friction, therefore should have the smallest pivot radius without exceeding the allowable unit contact pressures for the cup jewel and pivot material used.

Values of Constants in Equation (1) for Various Pivot Materials

Material	n	k	Material	n	k
Brass	0.245	153	Stellite	1.051	1.394
Steel (0.93C,0.1Mn) annealed	0.090	165	Stainless Steel (0.4 C, 12.5 Cr) quenched from 1,100 deg. C.		880
quenched from 780 deg. C.	0.408	1,215	quenched from 950 deg. C.		525
quenched from 800 deg. C.	0.405	1,247			

The design of V-jewel bearings such as shown in Figs. 3 and 7, involves the selection of the bearing cup radius and pivot radius. But the selection of these proportions depends on conditions existing over an area that is so small and so inaccessible that it is impossible to observe the bearing in action. Except for general performance tests, bearing proportions can be determined only from a theoretical analysis, and until recently, even such analytical methods were of little value.

Recent investigations at the National Physical Lab-

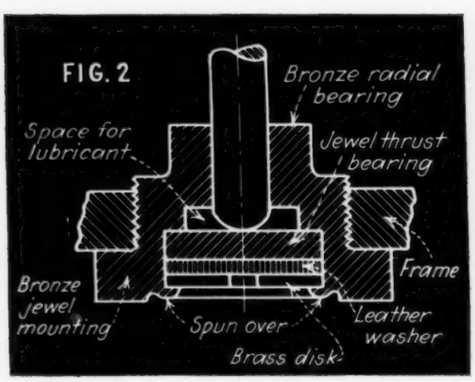


Fig. 2—Shown here in detail, the jewel bearing in Fig. 1 consists of a radial bearing with the jewel thrust plate spun in place

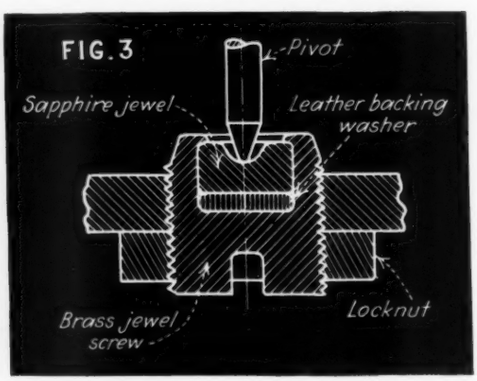


Fig. 3—In the V-jewel bearing, a hardened steel pivot with a spherical end bears against a spherical cup ground into the jewel

oratory show that the load supporting ability of microscopically sharp pivots increases with the cone angle. The relation between load per unit area p in dynes per sq. cm., and the half cone angle θ in deg., is given by the following equation determined empirically by Verney Stott in "An Investigation of Problems Relating to the Use of Pivots and Jewels in Instruments and Meters."

$$p = \frac{k}{1 + a \cot \theta} \qquad (1)$$

Values of the constants a and k in Equation (1) are given in the accompanying table, as determined by Stott from tests in which the load was sufficient to produce measurable flattening of the pivot end.

To determine the load per unit area p the radius of the contact area must be found. Referring to Fig. 4, where P is the total load in dynes, r_1 is the pivot radius and r_2 is the jewel radius, E_1 and E_2 are the elastic moduli for pivot and jewel, the radius a of the contact area in millimeters is given by:

$$a^3 = \frac{3P\ (E_1 + E_2)}{16\left(\frac{1}{r_1} + \frac{1}{r_2}\right)} \qquad (2)$$

For pivots with a spherical bearing surface the maximum load per unit area is approximately the same as

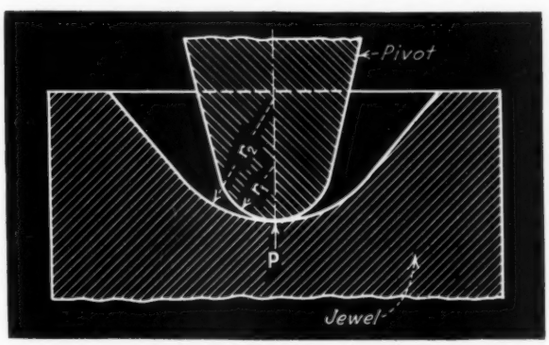


Fig. 4—Maximum unit bearing pressure is a function of the pivot radius, the jewel radius and the elastic constants of the bearing materials

for a pivot with a cone angle of 180 deg. since the radius of the contact area is small. The maximum load is therefore equal to k in the table. If this maximum pressure is not exceeded at the center of the contact area, permanent deformation will not occur. The minimum pivot radius R which will support this load k on a flat end-stone ($1/r_2 = 0$), is given by:

$$R = \frac{16\sqrt{3P}}{(2\pi k)^{1.5}\ (E_1 + E_2)} \qquad (3)$$

Although the elastic constants of sapphire have not been reported, it is assumed that E_2 is small compared with E_1 and that practicaliy all the deformation takes place on the pivot. The value of k for hardened high-carbon steel is approximately 1,250 kg. per sq. mm., but a value of 1,000 is used to be on the safe side. From these conditions the curve, Fig. 5, is obtained from Equation (3) giving the maximum load for a given pivot radius, or the minimum radius for a given load.

The curve applies only to high-carbon steel pivots on flat end-stones. If the pivot is used with a jewel radius of r_2 instead of a flat end-stone the corresponding pivot radius r_1 is:

$$1/r_1 = 1/R + 1/r_2 \qquad (4)$$

For other pivot materials given in the table, the pivot radius R given by Equation (3), is multiplied by the ratio of $k_s^{1.5}/k_o^{1.5}$ where k_s is for steel and k_o is the constant for the other material.

Having determined the minimum pivot radius for

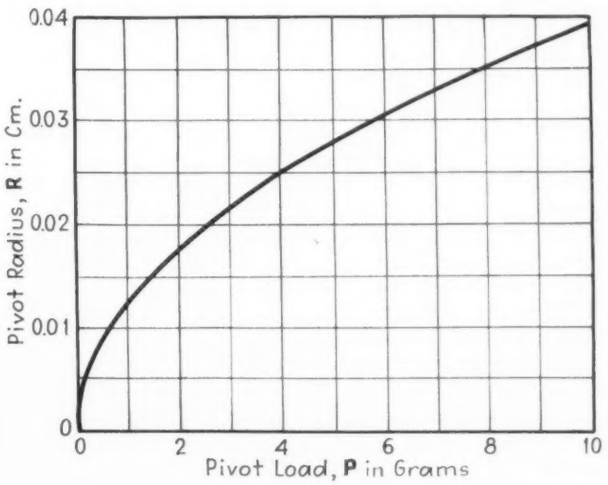


Fig. 5—Minimum pivot radius is plotted against maximum pivot load for a hardened steel pivot against a flat end-stone

the load, the frictional drag can then be calculated by the following method. Assuming that the unit pressure across the contact area varies from a maximum at the center to zero at the edge, Stott obtains a relation for the moment of the friction couple in terms of the coefficient of friction μ. For a pivot designed with the minimum radius for a given load according to Equation (3) the friction couple C is given by the following relation:

$$C = \frac{\pi \mu (3P)^{1.5}}{16\sqrt{2\pi k}} \qquad (5)$$

For hardened steel pivots and sapphire jewels, assuming a coefficient of friction of 0.15, Equation (5) can be plotted as in Fig. 6 to give the minimum friction torque for a given bearing load. The assumption of variable pressure across the contact area holds for pivot loads passing through the axis of the bearing cup. This condition is approached in vertical shafts carrying only their own weight, and for such designs Equation (5) indicates that the friction torque varies with the 1.5 power of the load, P. This relation offers

a ready comparison of the friction performance of designs having different weights.

Sensitivity, the ratio of the operating torque available for rotating the pivot to the friction torque, varies directly with the following relation between operating torque T, and the bearing load P.

$$K = T/P^{1.5} \qquad (6)$$

In this relation, K is the sensitivity, a factor of mechanical merit for designs wherein the loads are concentric with the axis of rotation. For shafts having

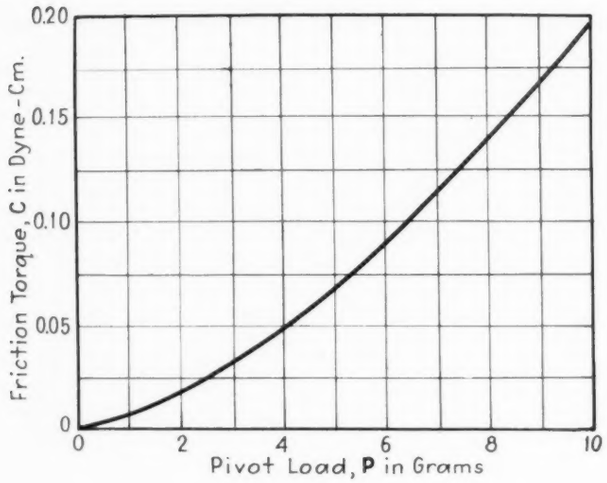


Fig. 6—Frictional torque for a given bearing load is given by this curve for steel-against-sapphire bearings. The curve applies only for pivots designed using Equation (3)

appreciable radial loads such as exist in horizontal shafts, the contact area and the axis of rotation do not coincide. Since the bearing must have considerable end play, the point of contact will be along the pivot flank. Under these conditions, the friction varies approximately in direct proportion to the load, and the exponent of P in Equation (6) approaches unity.

Materials selected for pivots are limited to those that are sufficiently hard and can be given a high polish. These two restrictions limit the choice of pivot materials to music-wire, hardened drill-rod or hardened stainless steel. Their inspection must be made by microscopic examination or by optical projection systems because of the small dimensions, and the high-grade polish that must be checked.

Sapphire, either natural or synthetic, is the most satisfactory material for small bearings. Sapphire possesses certain characteristics not found in other materials which make is particularly suitable for bearings in instruments. As explained in the previous article, there are materials that are less expensive than sapphire, but they are much less resistant to wear. There are also materials that are harder and equally resistant to wear, but more brittle and cause a greater wear of the pivot. For most jewel bearings, sapphire possesses a greater number of suitable characteristics than any other material.

Wear at the contact surfaces in jewel bearings is negligible, if safe unit pressures are not exceeded. However, friction has a tendency to increase during the life of the bearing because of the formation of rust in steel-against-sapphire bearings even when lubricated. This oxide formation has been explained by G. A. Tomlinson (*Proceedings of the Royal Society*, A Vol. 115, p. 472) as caused by molecular cohesion between the pivot and jewel at the point of contact. As the pivot rotates, sub-microscopic particles of iron are detached from the pivot, these particles combining with oxygen in the sapphire (Al_2O_3) to form iron oxide. This rust formation has been shown to depend to some extent upon the orientation of the crystal structure of the jewel, but there is little information available on this point.

Lubrication tends to flush the rust particles out of the contact area and therefore decreases friction. But the selection of a suitable lubricant, as outlined in the previous article, is a difficult problem, and in some bearings it may be desirable to use no oil.

The best data on life and wear of jewel bearings cover the bearings used in watt-hour meters. These bearings are of two general types as shown in Fig. 7. Service records of meters and inspection of replaced jewel bearings show that the most common failure of the pivot type bearing is the formation of a circular abrasion or roughening of the jewel surface. In other instances the pivot actually bores a new seat into the face of the jewel. Since neither of these defects is observed when sapphire bearings are replaced by diamond, the failures are undoubtedly caused by bearing pressures in excess of safe loads for sapphire, although defective jewels or excessive vibration or shock may also be contributing factors.

Another type of failure—held to be caused by methods used in polishing—is the appearance of small surface cracks or "riffs" in the polished jewel surface. These cracks are caused by internal strains in the jewel, and may appear at any time whether the bearing is in service or not. Depending on the type of bearing and the location of the riff these defects may or may not affect the bearing. For pivot bearings, riffs within the load-supporting area are undesirable if the shaft is vertical. With rolling contact riffs have no effect.

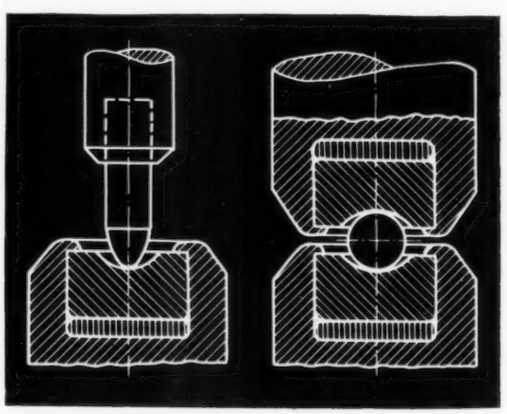


Fig. 7—Where there are no radial loads to support the bearing cups need be merely shallow spherical depressions. Both the pivot type (left) and the ball type with two jewels (right) are in general use

Electromagnets and Contacts for Magnetic Contactors

E. W. SEEGER
Assistant Chief Engineer, Cutler-Hammer, Inc.

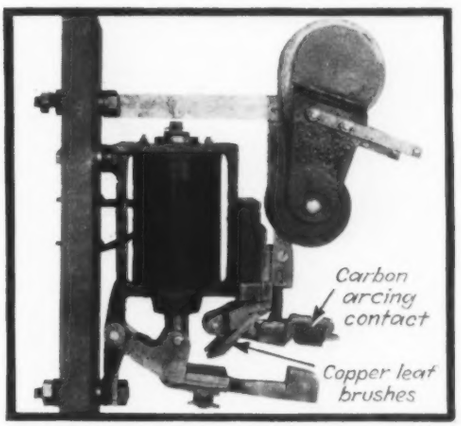
IN ORDER to make them adaptable to a wide range of motor control applications, magnetic contactors must have current carrying contacts capable of handling large amounts of power. Yet, these contacts must be operated by a relatively small amount of energy. For any given job the design of the magnetic contactor is governed by such factors as current, voltage, number of circuits and frequency of operation. In some applications high ambient temperature or other special atmospheric conditions also affect the design. In still other cases, limited space, ability to withstand shock, weight restrictions, or the desirability of using interchangeable standard parts are governing factors.

The first problem is to provide current-carrying contacts and current interrupting means to meet the requirements. For most applications the contacts must not only carry their rated current without exceeding a temperature of 65 deg. C., but must also be able to make and break a much higher current. The current carrying capacity in any given design depends primarily upon the size of the contacts and related parts, the material in the contacts, the contact pressure, and the number of the contact surfaces across which the current must pass.

Materials for Contacts

Copper, silver and carbon are the materials ordinarily used for contacts, copper being the most commonly used because of its low cost, good thermal conductivity, machinability, and long wear. On the other hand, copper oxidizes readily, the oxide increasing the contact resistance and causing overheating. Ordinarily, knife switches and circuit breakers, if left closed for extended periods, are susceptible to the same difficulties. For this reason, some type of wiping action is usually provided to keep the contact surfaces clean when the contactor is operated infrequently. The wiping contact is not favored for such service as for welding and steel mill equipment where frequent operation would cause undue wear. For such applications rolling or butt contact is preferable

Silver contacts are used on installations where operation is infrequent and where copper would be likely to oxidize and overheat. In some applications where space is a consideration, it is often possible to produce a smaller design by using silver.



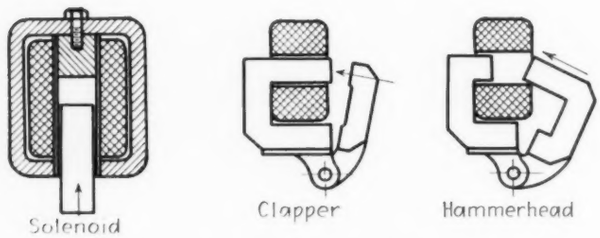
In interrupting the circuit, the copper-leaf brushes open in advance of the arcing contacts

Copper or silver contacts that are not properly maintained, especially on low voltage circuits, may weld together and fail to open the circuit when the operating coil circuit is broken. For this reason carbon contacts are used for elevator controls as a safety measure. However, carbon contacts are not recommended for ordinary service as the contact resistance varies, depending on how the contacts happen to seat. In general, the continuous capacity of a set of carbon to copper contacts is from 20 to 40 per cent of similar copper to copper contacts, depending on the grade of carbon used.

Contact pressure, one of the most important considerations in contactor design, influences the amount of current that can be carried without exceeding the allowable temperature rise. High contact pressure decreases contact resistance, increasing the ability of the contactor to close heavy currents. During the closing operation, the contacts move at a fairly high speed and tend to rebound, drawing an arc which causes rapid wear of the contact and increases the tendency to freeze. Rebound is reduced by using a high spring pressure in combination with contacts having low mass. But if the spring pressure is increased too much, a larger operating magnet will be required and the contactor will be unwieldy.

The depth of copper which can be worn away before the contacts fail to touch also has a considerable bearing on the size of the magnet. As the wear allowance is increased, the contact life is increased, but the magnet is loaded earlier in its stroke and a larger magnet is necessary.

Interrupting capacity, in addition to current carrying



Stronger pull is obtained from the solenoid magnet, but the hammerhead magnet combines the pull of the solenoid with the hinge motion of the clapper type

capacity is also an important factor in the design of a contactor. For heating or lighting circuits, the contactor is seldom required to interrupt a current larger than its rated capacity. For motor control, particularly when used in connection with squirrel-cage motors, a contactor must be capable of opening the stalled motor current which in some cases is as high as ten times the normal running current. For installations such as power plant auxiliaries, where a short circuit in one motor might cause other motors in the same group to shut down, a contactor must be capable of interrupting the short circuit current of the transformer to which it is connected.

For the same current, an inductive circuit is more difficult to open than a non-inductive circuit; high voltage presents a greater problem than low voltage; d.c. ruptures less easily than a.c.; and a low power factor load represents greater difficulty than a high power factor load.

Single break contacts with barriers placed between the poles are used for low voltage circuits where high interrupting capacity is not required. For greater interrupting capacity—for example, for squirrel-cage motors up to 7½ hp.—double-break contacts are used. For higher ratings, contactors are normally supplied with magnetic blowouts, forcing the arc upward, cooling and isolating it until the arc is extinguished. Arc horns, arc splitters and special shapes of arc chutes are used when necessary to increase the effectiveness of the blowout structure.

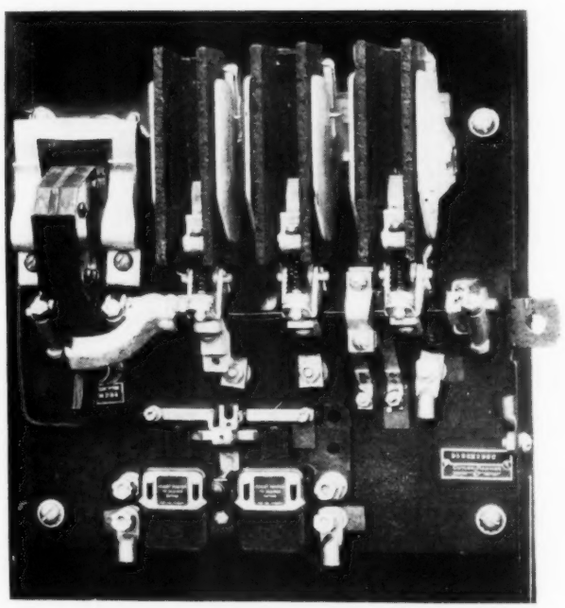
Types of Electromagnets

Oil-immersed contactors are used for high voltage circuits generally, and on some low voltage installations where corrosive atmospheres or an explosive hazard is present. Since magnetic blowouts are not effective under oil, the arc is extinguished by "stretching." Contact life in an oil-immersed contactor is usually no more than 10 to 15 per cent of the life on an equivalent air break design.

When the contact structure, wear allowance and contact pressure are determined for a given design, the appropriate characteristics required of the operating electromagnet can be determined from a load curve of the mechanism. The electromagnet must be designed to operate at voltages of 10 per cent or more below normal when the coil is at maximum operating temperature. Electromagnets used on d.c. usually have solid frames and armature levers of cast iron, cast steel, malleable or wrought iron or rolled steel. Electromagnets for a.c. are built up of punched laminations to reduce eddy current losses, except for very small structures for which it is more economical to use solid parts. Large d.c. electromagnets are also laminated when necessary to provide a higher speed of operation.

Electromagnets are of two general types: the solenoid or plunger type, in which a plunger moves inside the coil, and the clapper type, which has a pivoted armature attracted to the stationary poles when the coil is energized. The pull of an electromagnet is made up of two components, the solenoid effect and the pull caused by attraction. The solenoid effect is the pull on the plunger when placed inside the coil with the iron frame removed. With the clapper type of magnet no solenoid effect is obtained. But with the hammerhead magnet having a pivoted armature with the clapper extending into the coil, the pull obtained is from a combination of attraction and solenoid effects. Although more work per unit weight of material can be obtained from the plunger type magnet, the clapper type has mechanical advantages.

The most efficient coil is one in which the coil space is entirely filled with copper, as such a coil requires the least energy. This type of coil is not possible, as insulation must be provided between turns, between layers, and between the coil and the other parts of the structure. The covering on the wire (enamel, cotton, silk, asbestos) serves as insulation between turns and sometimes for insulation between layers. Coils are wound on insulating bobbins or are form wound and then taped to provide insulation between it and other parts of the structure.



Rolling contacts are closed by a laminated hammerhead electromagnet, the arc being extinguished by magnetic blowouts in this a.c. motor starter

Contactor coils are generally impregnated with an insulating compound which provides a more rigid structure and assists in preventing localized heating and the entrance of moisture.

Avoiding Common Defects In Single-Bearing Mountings

HERMANN HILL

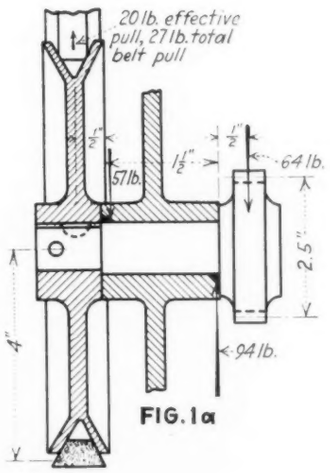
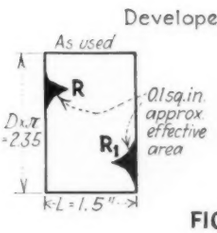


FIG. 1a



Developed Bearing

Based on 0.1 sq. in. effective bearing area at each of the reaction loads, R = 57 lb. and R_1 = 94 lb., the unit bearing pressures will be 570 lb. per sq. in. and 940 lb. per sq. in. respectively both of which are too high for lubrication

FIG. 1b

Fig. 1*a*—The bending moments created by belt pull and gear thrust are imposed on the single bearing. Fig. 1*b* shows the concentrated bearing pressures, which cause wear, resulting in the condition as indicated in Fig. 1*c*

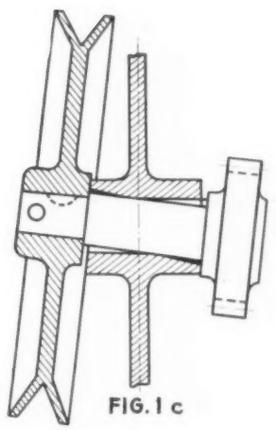


FIG. 1c

BECAUSE the bearing determines to a large extent the life of a machine, great importance is attached to its design, especially when the selling price of the machine is relatively high. But this is not always true for machines whose cost is comparatively low and that are sold in a highly competitive market, such as household appliances. Frequently in such designs an attempt is made to use the smallest possible number of bearings by providing only one bearing for a shaft instead of using the conventional two-bearing support. Such "single bearing" construction is inherently deficient except in instances where special provisions are made to satisfy the essential requirement of good bearing design, namely, to have the bearing pressure distributed practically uniformly along the full length of the bearing.

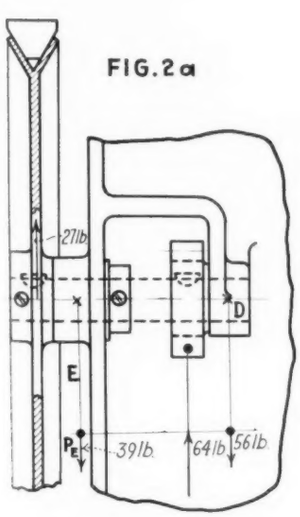


FIG. 2a

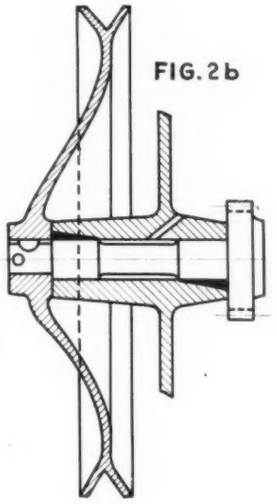


FIG. 2b

Fig. 2*a*—If there is insufficient room for two bearings, pulley hub can be offset as shown in Fig. 2*b*

A type of bearing assembly frequently seen in domestic machines is shown in Fig. 1. It consists of a shaft having a pinion mounted at one end and a pulley at the other, with a single bearing in between. For the sake of compactness and economy of material, the bearing is usually extremely short. The reason why such a bearing cannot render proper service is evident from the figure. With the loads concentrated at the ends of the bearing, the edges are soon worn away. Although this slightly increases the bearing area, the effect is not appreciable and heat is developed because the lubricant is squeezed out by the extreme pressure. An examination of such a shaft will always show the tell-tale marks, the blue rings where heat was generated at the contact between the shaft and the bearing metal. Often one will also find some of the bronze of the bushing deposited on the shaft, visible proof of the destructive action that has taken place.

As the bearing continues to wear, the contact area increases, but at the expense of the shaft fitting and alignment. Eventually the shaft becomes so loose that a new bushing is required, or reboring and bushing. If the bearing bushing is not replaced, the shaft will have considerable play in the bearing and will rock about the center portion of the bearing where no wear has taken place. In Fig. 1*c* it is shown how this condition develops.

In the description of a new design of domestic refrigerator recently brought on the market, a single

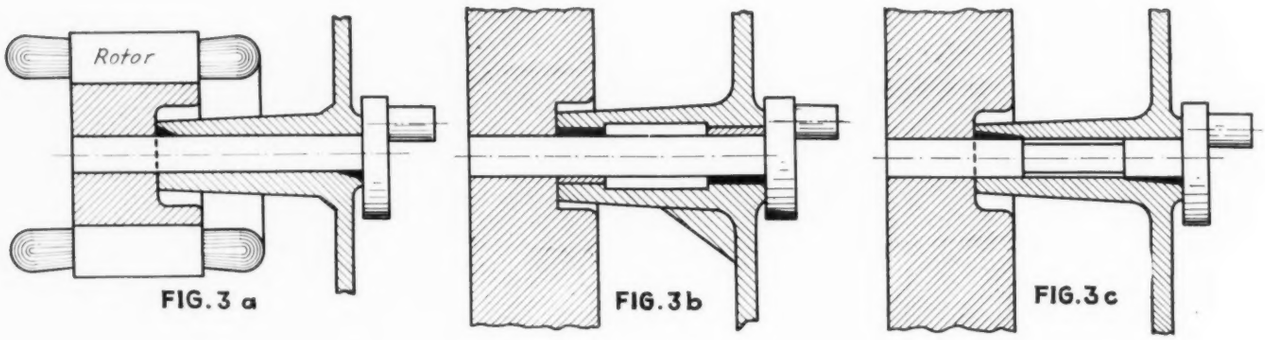


Fig. 3*a*—Using a long single bearing does not improve conditions greatly unless the bearing is counterbored as in Fig. 3*b* or the shaft is necked as in Fig. 3*c* in order to allow proper lubrication

bearing construction was emphasized as a special feature. In this particular design, which is illustrated in Fig. 3*a*, the bearing was made extra long, thereby reducing the reaction load at the end of the bearing. But although by lengthening the bearing the magnitude of the concentrated bearing pressures has been reduced, the pressure concentration is not eliminated and the center portion of the bearing does not serve any purpose. In fact, the center portion is detrimental in this particular design since it prevents the free flow of lubricant. Such a bearing can be improved greatly by providing a large clearance in the center portion of the bearing, which can be done in several ways. The bearing itself can be recessed or counterbored or two bushings, one at each end of the bore, as shown in Fig. 3*b* can be used. If this is not permissible, the shaft might possibly be turned to a smaller diameter for about one-third the length of the bearing, as shown in Fig. 3*c*. The recess thus formed serves as an oil well and aids in the distribution of the oil to the ends of the bearing. And while this type of bearing cannot possibly have perfect bearing pressure distribution, the distribution is greatly improved.

It will be noticed that in the design shown in Fig. 1, the belt pull and tooth load are opposed. This is the worst condition because the bending moments are in the same direction. The bearing will wear rapidly, the shaft will soon be out of alignment and the gear teeth will be worn tapered or possibly break on account of the concentrated tooth pressure at the end of the teeth. The only proper solution for such a condition is the addition of a second bearing, as shown in Fig. 2*a*. If it is not permissible to do this on account of lack of space, the single bearing design can be improved greatly by offsetting the pulley as shown in Fig. 2*b*, thereby putting the belt pull load directly over one end of the bearing.

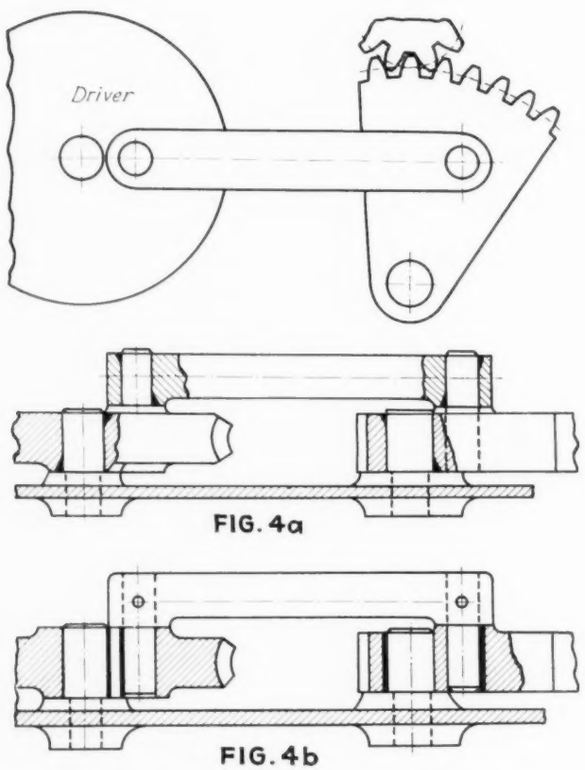


Fig. 4*a*—Here is shown a common defect in the design of crank mechanisms. By reversing the bearings as shown in Fig. 4*b* the concentrated bearing pressures are eliminated

Bearings for Crank Mechanism

In Fig. 4*a* is shown a typical design of crank mechanism wherein single bearings are used improperly. In this particular design, the whole assembly is operated in a bath of oil, but because of the high bearing pressures it was impossible for the lubricant to reach the bearings, as indicated by the blue rings around the shaft.

The faulty construction shown in Fig. 4*a* can be avoided readily by a few simple changes, as shown in Fig. 4*b*. The line of action of the loads on the bearings passes through the mid point of the bearing as indicated in the figure. This was accomplished by fixing the crankpins on the connecting rod which was designed sufficiently stiff to maintain the pins in parallel alignment. This simple change added practically nothing to the cost, the bearings have not shown any visible wear, and close inspection after test has not shown any discoloration of the crank pins because of concentrated pressures. This, however, is one of the few instances wherein it is possible to load the single bearing centrally.

Central loading of a single bearing can sometimes be accomplished by reversing or inverting the mechanism. Thus, in Fig. 5*a* is shown a single bearing construction

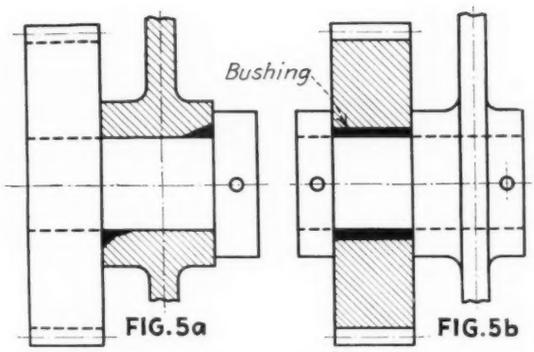


Fig. 5*a*—When using a stud shaft pressed in the wheel hub and supported in a single bushing, the bearing pressures are concentrated at the ends. In Fig. 5*b* is shown how central loading can be obtained by bushing the wheel

wherein the bearing carries the stud shaft of the gear or pinion. By making the stud shaft a fixed member and bushing the gear, as shown to the right of the figure, the single bearing is loaded centrally.

Another type of solution is illustrated in Fig. 6 wherein the construction is similar to the design shown in Fig. 2*b*. The rim of the gear has been offset so as to make the line of action of the gear tooth load fall inside of the single bearing.

If it is impossible to offset the rim of the gear or pulley, or bush the gear wheel, it becomes necessary to use a two-bearing type of design.

In Fig 7 is shown one example wherein it is impossible to use a single bearing construction that would be capable of giving satisfactory service. This represents perhaps the most severe type of bearing requirements. Even a relatively small amount of wear as indicated at Fig 7*b* will result in a tremendous increase in concentrated bearing pressures because of the added centrifugal forces created by the rotating cylinder. The only solution of such a problem is the addition of a bearing as shown at Fig. 7*c*, thereby eliminating the possibility of diagonal loading on the bearing and permitting proper lubrication.

In the design shown in Fig. 1, it will be noticed that the belt pull and gear thrust are shown as being opposite in direction. As stated, this is the worst condition. A better condition would be where the belt pull and gear thrust, or line of action of the gear tooth load, were equal and parallel and in the same direction. Under such a condition the resultant load, or the resultant of these two forces, would be parallel and midway between those acting on the wheels carried by the shaft. But only in rare instances will it be found that the two forces acting on each of the wheels mounted on the shaft will be equal. If they are not, the center of pressure on the bearing will be located near one of the wheels. It may be possible by rearranging the drives to procure a better pressure distribution. Thus by shifting the position of the motor or the point at which the gears mesh, it might be possible to greatly improve the bearing conditions. This, of course, also applies regardless of the type of wheels that may be mounted on the shaft, whether they be gears or pulleys.

Investigation of Bearing Failures

The result of investigations by the author as to the cause of bearing failures brought out the significant facts that in practically all instances the bearing that had failed was of sufficient size for the load which it was to carry. But in most instances, the failure was caused entirely by the fact that the load was not distributed over the full length of the bearing. Since it is rarely possible to design the mechanism so that a single bearing can be used with center loading, it is usually advisable to provide two bearings for each shaft, with the bearings so positioned that the resultant of the loads applied to the shaft will fall between the bearings. If the loads must of necessity be overhung, the distance between the two bearings must be large enough so that there will not be an excessive diagonal loading on the bearing. If a single bearing must be used, it should be made as long as possible to minimize the forces at the ends of the bearing and a recess should be provided in the center of the bearing to permit proper lubrication.

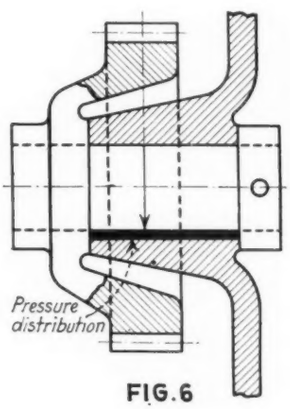


Fig. 6—Offset hubs will load the shaft bushing centrally, thereby utilizing the maximum bearing area and giving good conditions for lubrication

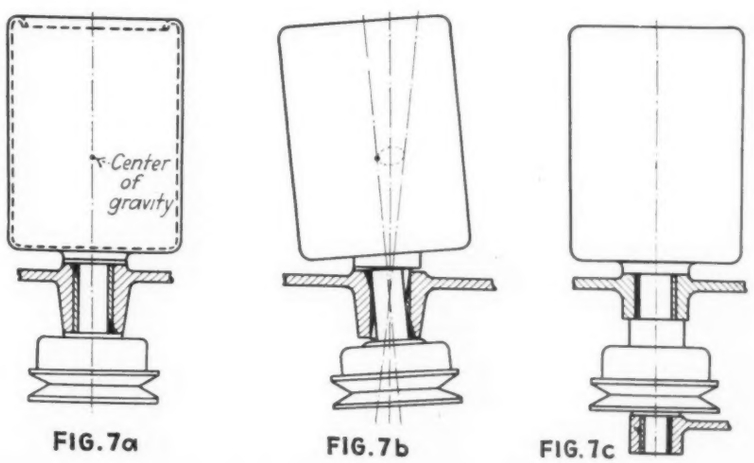


Fig. 7*a*—Where an overhanging rotating mass develops centrifugal forces, a single bearing will quickly wear as shown in Fig. 7*b*, thereby throwing the rotating mass out of balance. The only solution is to use two bearings as shown in Fig. 7*c*, with the pulley between the bearings

COMMUNICATION and COMMENT

Oil Circulated by Centrifugal Action

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Chief Engineer, Motorstokor Corporation

In a domestic coal burner a small, noiseless idler was required for a V-belt drive operating approximately 2,000 hours per year. At the same time it was necessary to provide a lubricating system that would not require frequent oiling. To meet these requirements the idler shown in the accompanying diagram was designed.

As shown in the diagram, a pivoted arm supports the axle on which the idler rides. This pivoted arm is free to rotate so that the whole assembly can swing and float on the belt.

The idler has an enlarged hub and a cap, both of which are die-castings. A bronze bushing, die-cast as an insert in the hub, is held on the axle by a thin disk attached to the overhanging end of the axle by a machine screw. This disk has a small drilled hole running from its periphery to a point in line with the bearing surface of the axle. At this point, it meets another drilled hole communicating with a channel formed by the beveled edge of the bronze bushing.

From this bevel, there is a helical groove in the bushing which terminates in an oil chamber within the right end of the bushing, from which there are four holes leading back into the pulley body. A labyrinth seal in the bushing prevents oil from leaking out of the pulley.

By removing the threaded plug the body of the idler is filled with a medium grade motor oil to the level of the bottom of the filling hole. When the idler turns, this oil is thrown to the outside of the oil chamber by centrifugal force, the pressure built up thereby causing the oil to flow into the radial hole in the stationary disk. The oil flows through the bevel and the spiral groove into the oil chamber, and is thrown by centrifugal force through the holes back into the main reservoir.

This design embodies centrifugal-pump action rather than pitot-tube action, for which reason the disk can be mounted in any direction and the idler works equally as well with either direction of rotation.

Idler pulleys of this design have been in use for more than four years, running at high speed without noise for an average of 2,000 hours a year. In many instances these pulleys have never been re-oiled after originally installed.

Managing the Engineer

M. G. DEMOUGEOT
Assistant Factory Manager
International Banding Machine Company

Why is it that so many otherwise very capable engineers or inventors complete the design of a machine or other item before working out the most important factors or accessories?

The example I have in mind is where a competent engineer developed a small printing machine for a special purpose. The machine was developed in a reasonable time and functioned satisfactorily. But even when ten sample machines were ready to be tried out in the field, the engineer had not found the special ink which was known beforehand to be needed, nor had he studied the conditions under which this machine was to be operated.

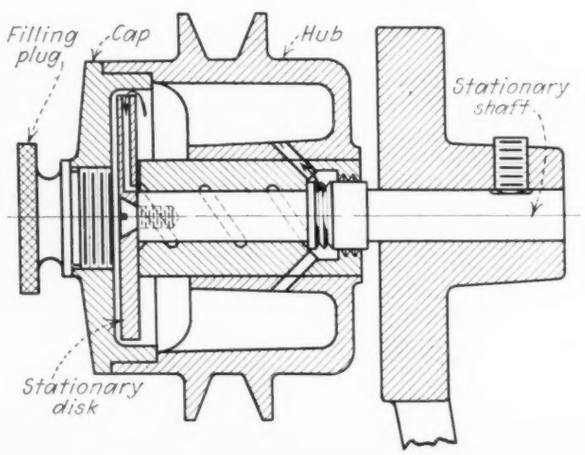
Prior to designing the machine he learned that certain inks could not be used and that undoubtedly an ink of a certain base would have to be used. He also knew exactly where the machine would be used, as well as the class of worker who would operate it. But he had not studied either point carefully.

He did not know the characteristics of the special ink required. And in a printing machine this is a vital factor, as it involves the feeding and application of the ink.

As it happened in this case, neither neglect proved very costly. They did however cause difficulty and delay in the production of this product. This man had repeatedly proved himself a competent engineer.

An engineer or inventor sees a machine in a different light than that obtained in the place where it is to be used. I believe it rests with the management to point out to the engineer one or more factors which must be kept in mind to the last detail when designing the product. The manager, or whoever is directing the engineer, should be capable of sizing up a new machine with an open mind, considering it from every angle, and not look at it simply as an engineering development or as an additional product to be made and sold.

In a medium or small-sized plant a proposed new product should not be placed at once in the hands of an en-



Centrifugal-pump action forces oil through the hole in the stationary disk into the helical groove in the bushing, and out into the pulley body at the right

gineer or inventor. It should be considered and investigated by one or more of the management executives and a list of factors, or points to bear in mind, made up for the engineer to consider when designing the product. Much time and money may be saved through proper forethought and consideration of all factors entering into the successful merchandising of a new product.

Calculus Not a Memory Stunt

To the Editor:

The engineering profession will be called upon in the future for more penetrating analyses which can be accomplished best through calculus procedure. The need for a more generally distributed ability to handle vibration, heat transfer, electrical and economical distribution problems will be needed. Calculus procedure applied to engineering problems not only provides a terminal result, but presents an intimate pattern of the relationships involved.

From four years' experience in teaching calculus at the John Huntington Polytechnic Institute, we find that the large majority of men come with the notion that calculus is a species of memorized ritual. Even those with previous instructions seem to feel that if enough is memorized, current engineering problems can thereby be automatically solved.

Calculus as a point of view is seldom acquired through the conventional text in present day instructions. While there are other ways of presenting the subject, we instruct through the development of mathematical patterns of integrals and derivatives, using the idea that any mathematical expression is simply a species of gearing.

It is not the profundity but the algebraic complexity of many desirable solutions which makes calculus solutions difficult or impossible.

—J. M. Barr
President, The Integraph Company

Design Formula for a Pin and Shaft of Equal Strength

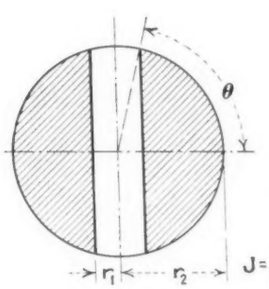
H. F. Bariffi
Hamden, Conn.

For all parts to be equally strong, the diameter of a pin passing through the hub of a pulley or gear bears a definite relation to the power transmitted by the shaft. Likewise, when designing safety shear pins, it is necessary to determine the maximum pin diameter that will permit the pin to shear before damaging the shaft. The following analysis gives a method for determining the relation between pin and shaft dimensions.

To simplify calculations, the following assumptions are made: (1) Shaft and pin are the same material; (2) Shear stress law for a circular section applies to drilled shaft; (3) No bending loads on shaft; (4) Both pin and shaft are excellent fits; (5) Shear area of pin is that of a normal section.

The last assumption is made to avoid the evaluation of an expression for the elliptical surface of the pin and the approximation introduces little error.

To meet the conditions of the problem, the torque exerted by the pin must be equal to the torque on a shaft. The pin torque T_p is given by Equation (1), in the accompanying diagram, where S_p is the shear stress in the pin. The shaft torque T_s is given by Equation (2), where S_s is the shear stress in the shaft.



$$T_p = 2\pi r_1^2 r_2 S_p \quad (1)$$

$$T_s = J S_s / r_2 \quad (2)$$

$$r_1 = \frac{1}{\sqrt{2\pi r_2}} \sqrt{J} \quad (3)$$

$$J = \left[r_2^4 \theta - r_1^4 \left(\frac{\sin\theta}{3\cos^3\theta} + \frac{2}{3}\tan\theta \right) \right] \quad (4)$$

$$r_1 = \frac{1}{\sqrt{2\pi r_2}} \left[r_2^4 \theta - r_1^4 \left(\frac{\sin\theta}{3\cos^3\theta} + \frac{2}{3}\tan\theta \right) \right] \quad (5)$$

Fig. 1—Proportions of a pin as strong as the shaft can be obtained from Equation (5) by a cut-and-try solution

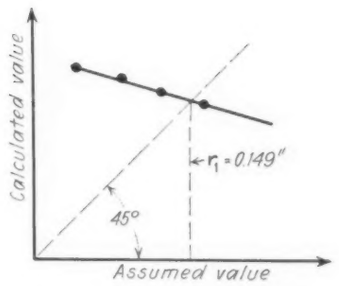


Fig. 2—Plotting calculated values against assumed values leads to a close approximation for a solution to Equation (5)

Equating the two torques gives Equation (3), in which the radius r_1 of the pin is expressed in terms of the shaft radius r_2 and J the polar moment of the cross-section through the axis of the pin bore. By an integration process the polar moment J is expressed in terms of r_1, r_2, and the angle θ in radians shown in Fig. 1, using the form given in Equation (4). Substituting this expression for J in Equation (3) gives the final Equation (5), which can be solved by cut-and-try methods.

To illustrate the use of this method, a shaft $\frac{3}{4}$ in. in diameter is to be provided with a pin of equal strength and of the same material. As a first assumption, r_1 is taken to be 0.0625 in., and θ is therefore 1.403 radians. Substitution of these values in the right-hand member of Equation (5) gives a value of 0.1732 in. for r_1. But r_1 was assumed to be 0.0625 in., showing that the first estimate is too small. Trying several other dimensions, a table of assumed and calculated values is obtained. Plotting the assumed values against the calculated values, using equal scales, will give a straight line shown in Fig. 2. A 45-deg. line from the origin intersects the plotted line at the point of the required radius. In this example, the radius of a pin having shear strength equal to the shaft is 0.149 in.

Wood Shrinkage

John E. Hyler
Peoria, Ill.

In its ordinary form wood is unsuitable for many uses because of the effect of humidity in causing shrinkage or swelling. However, these trou-

bles are largely eliminated in plywood for the reason that there is no shrinking or swelling in the direction of the grain. In building up plywood, the grain in one ply is laid at right angles to that in adjacent plies, making the plywood practically inert to humidity and therefore applicable to many places where a single thickness of wood would be undesirable. Metal-faced plywood is also considerably used, and a process for vulcanizing rubber to plywood has been developed, although this process is not yet commercially available.

There are places where shrinking and swelling characteristics of wood are used to advantage. In a hygrostat used to control humidifying apparatus, varying humidity in the surrounding air causes dimensional changes in a cylinder of wood, operating an electric switch.

Drafting Room Efficiency

To the Editor:

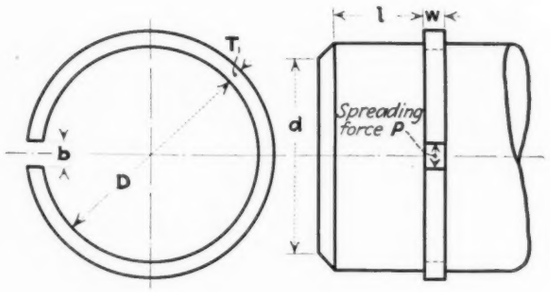
Because of the varying type of work, it is impossible to apply a piece-work system in most drafting rooms. It is only where work becomes highly standardized that some sort of piecework system could be applied. For construction or field drawings, there is no hope for any standardized classification. Depending upon the particular drafting force and regardless of the kind of work, the amount of detail required in drawings will give hourly variation of as much as 100 per cent.

For example, general architectural plans require very little time per drawing, structural drawings and foundations are average, while piping or other closely detailed work requires a greater number of hours per drawing. Also, it is impossible to evaluate the conception of the work itself. Certain projects will go through a drafting room very smoothly because the general outlines and underlying conditions have been carefully worked out before drafting work has been started. In other cases, because of incomplete preparation or other reasons such as economic changes and market changes, revisions in drawing are required to an extent often amounting to 100 per cent extra time over what would have been required had the work proceeded without interruptions.

For these reasons no system will be satisfactory which attempts to divide drawings into a number of classes, based on size, content and character of workmanship. If the supervisor should attempt to guess the "class" of the drawing, he must combine this guess with an equally important guess as to the number of drawings required for the proposed work. His chances of obtaining a close estimate by this method, are small.

The performance of the individual draftsman is subject to the judgment of the supervisor. There is no fairer way to judge a draftsman than to obtain a composite rating from several squad bosses for whom the man has worked.

—E. H. BANTON



Snap ring design involves several factors that are not subject to systematic analysis or calculation

Snap Ring Design

ROLAND V. HUTCHINSON
Lansing, Mich.

Variations in handling and abuse so circumscribe the design of snap rings that no general solution of the problem is possible. The important circumstances to be considered in any analysis of assembly stresses are details of the assembly method. The snap ring may be placed on a shaft by any one of a number of different methods. The ring may be opened up by placing a block in the gap, and the block removed after the ring is slipped into position, or reversed pliers with limit stops may be used to spread the ring. The end of the shaft may be chamfered as shown in the accompanying figure to a diameter d somewhat smaller than the free bore D of the ring and the ring shoved on by a collar; or the ring can be assembled by locally applied loads around the ring. Still other conditions exist when the ring is pushed on a shaft having no chamfer on the end.

If the distance l from the groove to the end of the shaft is small the ring can be twisted and partially positioned, the balance of the assembly operation completed by a combination of bending and twisting.

The general method of removing rings is to pry the ends apart with a screw driver and slip strips of sheet metal under the deformed ring, working the strips away from the gap. Other strips are worked under the ring, and it is finally removed by sliding over the strips and off the shaft.

In general, rings are more likely to be overstressed in removal than in assembly since, in initial assembly good handling control is possible, and the stressing of the ring varies with the facilities provided and the judgment of the workmen. It is therefore reasonable to find that "all engineers seem to have a different method for solving such problems," for all may not have had similar experiences with even the same design.

In setting up a mathematical solution for the problem, there are many unknowns; deflection of the ring, directions and points of application of deforming forces, intensity of deforming force which may be variable, as well as T, the thickness of the ring The situation is further complicated by the fact that the deformations encountered are large and that the ring is curved.

In even an approximate solution, deflections will involve the moment of inertia of the cross section as a multiplying constant, and stresses will involve the section modulus. These are functions of T^3 and T^2. Assuming that the neutral axis in bending is the locus of the centroids of the radial sections of the ring, this locus has an initial curvature of $\frac{1}{2}(2D + T)$. This factor enters into the expression for bending moment. The deflections also vary in some manner with the spreading force P. The determination of T will involve the solution of two simultaneous equations, one of which will contain terms including T^6, T and P. The other equation will contain terms including T^2 and P.

But these equations will be of little value since conditions of manipulation may differ greatly from those assumed at the start of the analysis. For this reason, the use of the snap ring should be avoided where possible. A sub-

stitute frequently used is a horse-shoe half-washer, radially restrained by a counterbore in the hub of a member on the shaft.

Where the use of snap ring is necessary, the best procedure is to make samples of different proportions, and to assemble and dismantle them with as close an approach as possible to expected field conditions. The variations between formulas used by different engineers are evidence of variations in the experiences of the formula makers.

Did You Get Your Extra Copies?

To the Editor:

At your convenience we will be pleased to receive from you twelve copies of "Inquiry Form for Electric Motors," as shown on page 145 of the April number of *Product Engineering*.

—B. E. S.

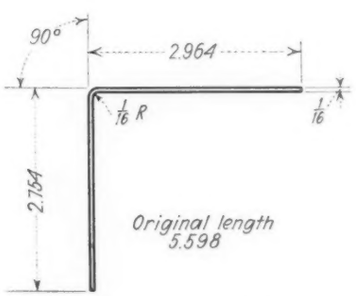
Editor's Note—This letter is only one of numerous requests we have received for copies of "Inquiry Form for Electric Motors." A limited number of these forms, printed on bond paper are still available, and will be sent free of charge to readers upon request. With your inquiry please indicate the number of copies you need.

Bending Allowances

To the Editor:

Applying two methods for computing the developed length of metal blanks, the calculated allowances do not check each other. In an experiment a ⅛ in. thick, 17 S.T. aluminum alloy plate 5.598 in. long was bent as shown in the diagram. Calculating the original length from the final dimensions by the National Cash Register method:

$$L = A + B + S\,(0.40\,t + R)$$



Methods for determining bending allowance do not agree with actual dimensions for this aluminum alloy angle

$S = 0.4,\ t = 0.0625,\ R = 0.0625$
Side B $= 2.964 - 0.125 = 2.839$
Side A $= 2.754 - 0.125 = 2.629$
Allowance $S(0.4\,t + R) = 0.035$
Original Length, L $= 5.503$

This length is 0.095 in. shorter than the actual dimensions. Using a second method, which appeared in the *American Machinist*, Nov. 4, 1926:

$L = A + B - m$
Side A $= 2.964$
Side B $= 2.754$
5.718
Less Allowance m $= 0.104$ (from chart)
Original Length, L $= 5.614$

From the second method the calculated length of 5.614 in. is 0.016 in. shorter than actual. Thus, the second method gave results considerably more accurate than results from the first one.

What is the reason for the difference between these two methods and the results obtained from them?

—F. L. Tuxbury
Oakmont, Pa.

Spiral Curves for Fillets

To the Editor:

Fillet curves proposed by R. V. Baud in his article "Fillet Profiles of Constant Stress" appearing on page 133 of the April number, are new and will be of assistance in eliminating stress concentration at the throat of the fillet. These semi-theoretical curves will eliminate concentrated stresses at the point where the fillet joins the smaller section, but a localized stress will be produced at the abrupt change in the boundary line where the fillet meets the larger section.

Thus, the proposed curves shift the point of stress concentration from the throat of the fillet to the lower extremity, and because of the abrupt change in boundary, the new curves may introduce even larger stresses than are present in the circular fillet.

In a photo-elastic investigation at Columbia University two years ago made for the Westinghouse Electric and Manufacturing Company, a parabolic spiral curve was developed to eliminate stress concentration at the root of the fillet. This curve is given by the formula:

$$\rho = k\,\theta^{0.5}$$

This spiral curve can be readily adapted to fillet design, once a set of curves are drawn to different scales. With several standard spirals available, the one that best fits the fillet space can be quickly selected and specified.

—A. G. Solakian
Columbia University

Volume for Intersecting Cylinders

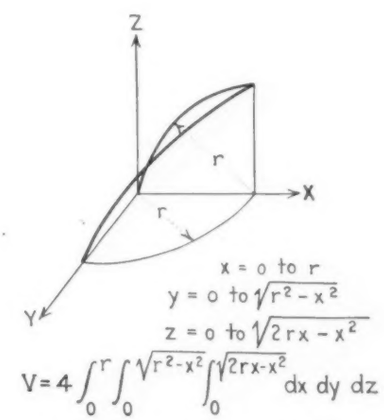
To the Editor:

In the February number, page 70, Floyd Carlson requested a formula giving the volume removed from a solid column to allow a pipe of the same diameter to cross at right angles to the column. The depth of interference was equal to the radius.

The volume V of any solid bounded by a surface whose equation is given between three rectangular coordinates x, y and z is given by:

$$V = \int\int\int dx\,dy\,dz$$

In this problem, the limits of x, y and z are as shown in the accompanying diagram. Since this volume is only ¼ of the total, the entire intercepted volume will be given by the formula for V in the diagram. Expressed in terms of the radius r, the required volume is $2.325\ r^3$ for this problem.



Triple integration between the limits shown gives the total volume intercepted by the two intersecting cylinders

—Manious Gottlieb
Philadelphia, Pa.

Errata

In the Reference Book Sheet appearing on page 119 in the March number, the constant in the deflection formula was given incorrectly as 0.35. The correct value for this constant is 0.035.

Also, in the leading article of the April issue two errors escaped the proofreader. Henry Dreyfuss, designer of the General Electric refrigerator and the Todd Protectograph, was listed in two captions as Walter Dreyfuss, and Russel Wright's first name was misspelled with two l's. We offer apologies to both gentlemen.

Typical Designs of Lock Bolts

ADAM FREDERICKS

Fig. 16—A compact design of lock bolt mechanism wherein the spring backed bolt *A*, shown in engaged position, is released by hand-operated cam *B*. The engaging lug A_1 is held in correct relation to the cam by the dog-pointed-set-screw *C*, engaging the keyway in the bolt.

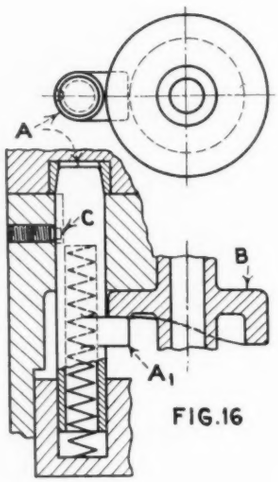


FIG.16

Fig. 17—A patented positioning device used on the Bullard vertical turret lathe. Two locking rings *A* and *B*, each having a series of five projections or bolts, engage one another. As each bolt and space occupy 36 deg. of a circle, exact finishing is required, the grinding limit being 0.0001 in

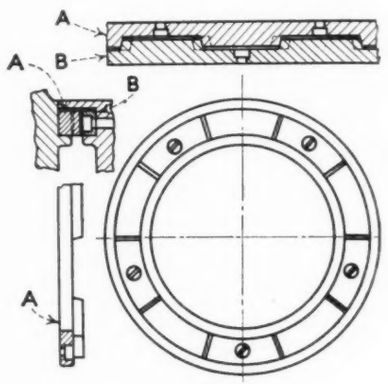


FIG.17

Fig. 18—The hardened and ground cylindrical lock bolt *A* is guided by the taper pin *B* and actuated automatically by lever *C* which carries a roller *D* that bears against the lower front end of the bolt. The side walls of the grooves in lock bolt plate *E* are local hardened.

Fig. 19—In this design plates *A* and *C* are fastened together, the former accommodating the bolt *B* while plate *C* is positioned by bolt *D*. Rotated in the direction of the arrow, bolt *B* slides into the slot in plate *A*, one side being milled to 20 deg. When indexing begins and bolt *D* is pulled away, a 45 deg. slot in plate *A* pushes out the bolt *B*, both bolts then riding on the periphery of their respective plates, and bolt *D* sliding down the easy incline *F* to a predetermined depth.

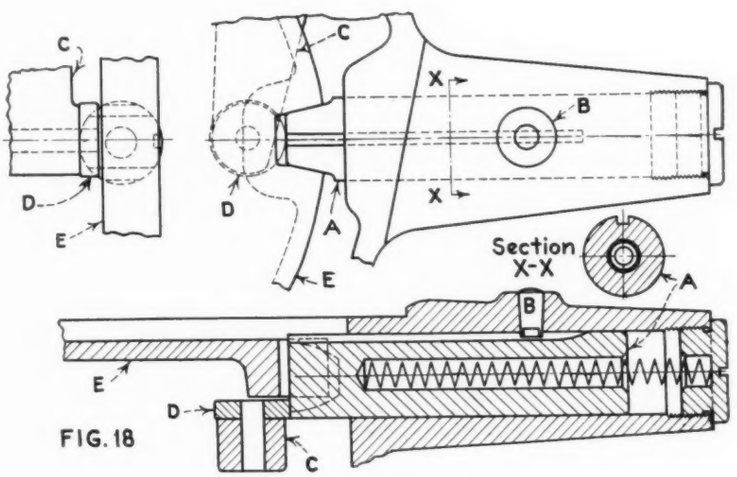


FIG. 18

Fig. 20—Bolt *A*, integral with lever arm A_1, has ears *B* and *B* directly above the pivot shaft at B_1, fitting on either side of the plate *C*. As the plate reciprocates it pulls the bolt along with it. Cam *D* contacts the roller *E* in lever A_1, the cam being long enough at *X* to accommodate the required travel of plate and bolt. The welded ring *F* fits in the groove in the hub of plate *C* and is connected to tension spring *G*, the other end of which engages a pin in lever A_1.

Fig. 21—Tension spring A is used to keep the bolt in engagement. Such a spring must necessarily be stiffer than a compression spring. Its disadvantage lies in the offset of the spring. especially when the lock bolt plate is in the position as shown at the left.

Fig. 22—If space is limited, a swing bolt may be used. Bolt *A* is pivoted at *B* and operated by hand with cam *C*. At the end of the cam *C*, and

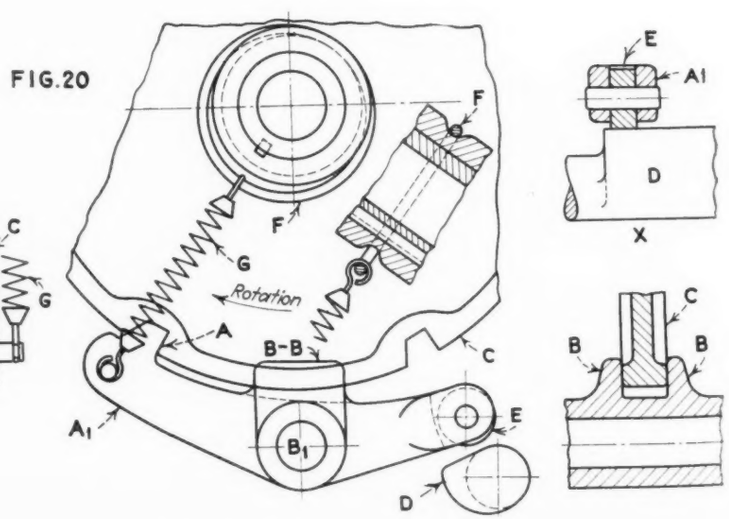


FIG.20

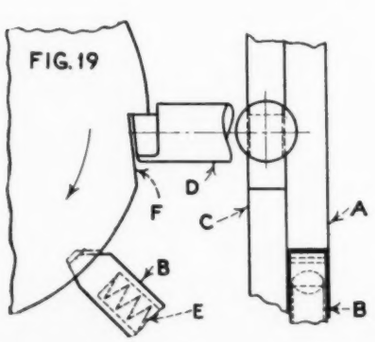


FIG.19

for Indexing Mechanisms—II

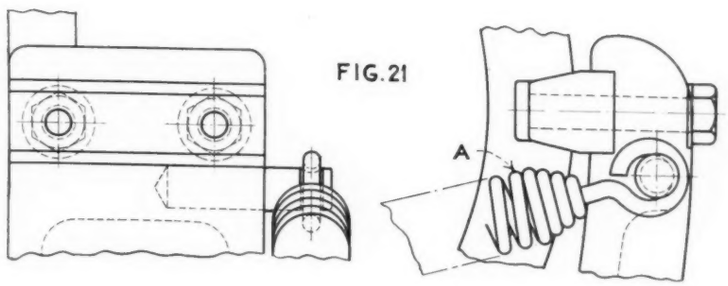


FIG. 21

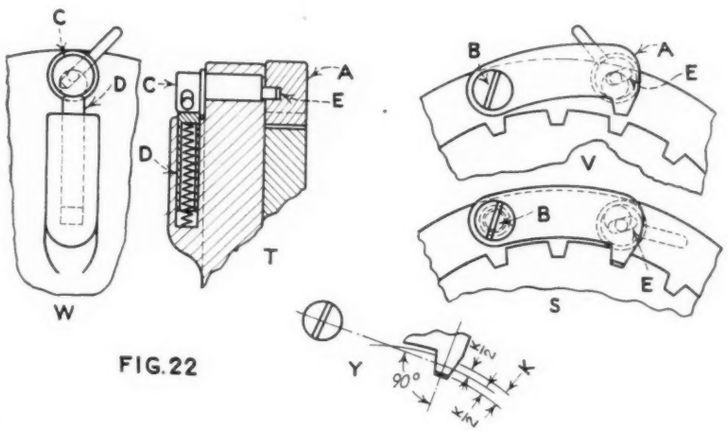


FIG. 22

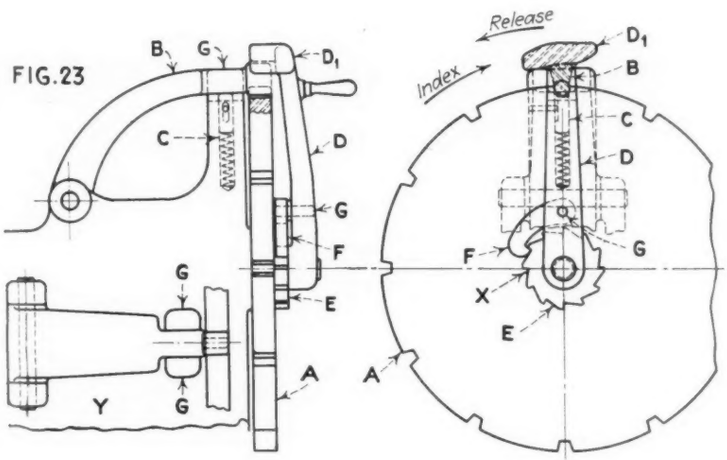


FIG. 23

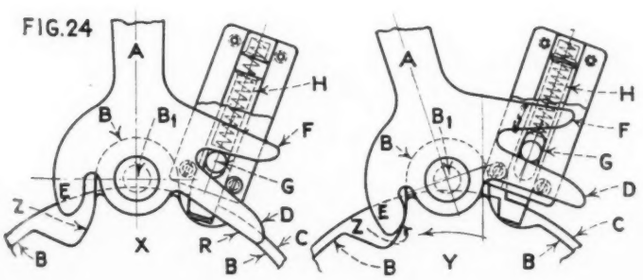


FIG. 24

integral with it, is an eccentrically turned projection engaging slot E in bolt A. At S and T the bolt is engaged, at V and W disengaged. For accuracy, the center of the pivot must be located as shown at Y, where K indicates depth of engagement.

Fig. 23—By using a lock bolt plate A many times larger than the work, the indexing error is diminished. The swinging lock bolt B is released automatically by spring plunger C, which has a predetermined movement, when the hand operated lever D is moved to the left as shown by arrow marked *"Release,"* Cam D_1 contacts the rounded top surface of lock bolt B. Ratchet E is keyed with the lock bolt plate A to the spindle. As the lock bolt is released and lever D is rotated 30 deg. counter-clockwise, pawl F engages the next tooth in the ratchet wheel at X. Lever D is then pulled in direction of arrow marked *"Index,"* the cam D_1 moving the lock bolt downward into the next opening in the lock bolt plate. At Y is a plan view of the bolt showing the two centering lugs G and G between which the lock bolt is additionally supported.

Fig. 24—Handle A is mounted on the plate B at B_1, and is independent of the lock bolt plate C. As handle A is pulled to the left, as shown in view Y, prong D pushes against pin G driven into the spring backed lock bolt H, thereby disengaging the bolt. At the same time prong E contacts against the plate B at Z. Both plates then move simultaneously, releasing lock bolt H which rides on the periphery of the lock bolt plate C until the bolt falls into the next slot. Handle H is then pushed back again, clockwise, prong D contacting plate B at R upon which prong F pushes against pin G, seating the lock bolt in a locked position.

Fig. 25—In this design the plate is indexed through a half revolution in one direction and then back again in the opposite. The lock bolt plate A is keyed to the spindle. The lock bolt operating plate B is free to rotate on the spindle. View on the left shows the locked position. When indexing handle C is pushed counter-clockwise, as shown at the right, groove F in plate B forces the lock bolt D out of engagement. Pin G, driven into plate B engages slot H in plate A, thereby lining up groove J with slot K. Upon further movement in a counter-clockwise direction, the roller E on the bolt D may slide into groove J and the bolt enter slot K. The dot and dash line in both views show the positions when indexing in the opposite directions.

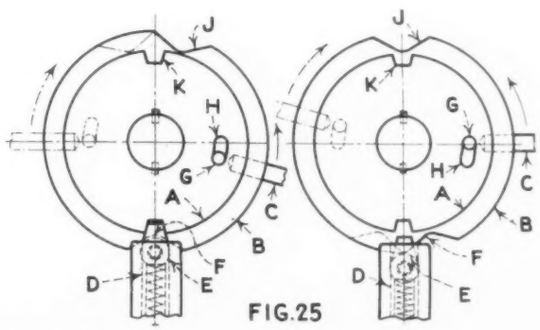


FIG. 25

· NEWS ·

Diesel Powered Streamlined Train Exceeds 100 M.P.H. in First Run

In a run from the plant of the Edward G. Budd Manufacturing Company, in Philadelphia, where it was built, to Mansville, N. J. and return, the Burlington "Zephyr" glided over the tracks of the Reading Railroad at a speed of 104 miles per hour. So smoothly did this train of shining shot-welded stainless steel run that the invited press representatives could not distinguish whether the speed was 75 m.p.h., or 100 m.p.h., except by the evident increased rapidity with which telegraph poles and mile posts flashed by. Comfortably seated in the adjustable-back aluminum-frame upholstered chairs, there was no noticeable side-swaying and jolting.

In many respects the same general type of design was followed as was used in the rail cars and trains built previously by the Edward G. Budd Manufacturing Company. To the casual observer the principal features are the sealed windows mounted in either permanently-plastic putty, or in rubber strips; thermostatically controlled air-conditioning; indirect lighting and the interior finish relative to window curtains, draperies, upholstery and finish.

Underframe and body of this three-car articulated train is built entirely of stainless steel fabricated from sheets and shot-welded by the patented Budd process. A monocoque construction is used, the skin being said to carry about 70 per cent of the load stresses. For this reason, the stainless steel roof sheets have square corrugation, running lengthwise, a cross-section of the sheet appearing much the same as the profile of a square-thread screw. Similarly, the side sheets below the windows are fluted. All of the sheathing except that at the front end is 0.030 in. thick. Corrugated sheathing is also used below the underframe, to inclose all of the tanks, piping and brake-rigging, thereby eliminating much wind resistance. Hair-felt padding cemented to the inside of the sheathing serves to further insulate the cars against noises.

Aluminum foil is used for insulating the walls and roof of the cars against heat. Additional insulation is provided in the passenger compartment by finishing the side-walls with Masonite panels and using 5/16 in. cork tiling for the floors. This tiling is cemented over the corrugated steel sheet floor, the spaces in the corrugation being first filled with ground cork. Wood flooring is used in the mail compartment and in the baggage compartment.

One of the outstanding features of this stainless steel shot-welded train is the Winton 600 hp. high-compression, two-cycle, eight-in-line diesel engine. Running at 720 r.p.m., it is direct-connected to the main generator through a flexible steel-disk coupling. Power from the generator is furnished to two General Electric series wound traction motors mounted on the front truck and which drive the axles through a single gear reduction.

Although the trucks are of a standard type, they have many unusual features. The axles are hollow-bored, not only to save weight, but also to make possible improved heat-treatment of the axles. Timken roller bearings are used on all the axles, and the motor armatures are mounted in SKF roller bearings. The use of these bearings reduced the frictional resistance to such a point that ten husky men can pull the train with a rope, as was demonstrated to the representatives of the press.

Rubber insulation is used extensively in the trucks in order to minimize noise and vibration. Rubber inserts are used under the center plate, at the top and bottom spring seats, at the top and bottom seats of the equalizer springs, and where the equalizers are supported at the journal boxes. Rubber pads vulcanized to Oilite metal plates are used for liners in the pedestals and similar plates are used for insulation between the bolster and the truck frame.

To procure every possible weight saving, aluminum is used extensively for parts in the air brake equipment that do not require great strength. The air compressor frame is made of aluminum. Similarly, to reduce weight to a minimum, the evaporator units in the air-conditioning equipment are made of aluminum, which in this application also has the advantage of having a high rate of heat conduction. The compressed-air reservoirs are made of stainless steel.

Except for a few special fittings, all piping for water, steam and air is of copper, with sweated fittings. It is estimated that the use of copper made possible a weight reduction of as much as 60 per cent.

After being turned over to the Chi-

America's first Diesel all-steel streamlined train, the Burlington "Zephyr," built by Budd. It is 196 ft. long, weighs 195,000 lb., seats 72 passengers, and can carry 50,000 lb. of baggage and express. In the first car is the operator's cab, engine room, railway post office and mail storage space, the second car has a baggage and express compartment, buffet-grill and 20-passenger compartment. The third car has a 40-passenger compartment and a 12-passenger observation lounge

cago-Burlington & Quincy on April 18, the train started on a ten weeks' tour of the principal cities of the country, after which it will be placed on exhibition at the Century of Progress Fair in Chicago. When finally put into regular railroad service it will be run between Kansas City, Mo., St. Joseph, Omaha, Nebr., and Lincoln, it being expected to make this 251 mile run on a four-hour schedule.

TVA Demands Low Cost Appliances

As announced recently in the press, the Tennessee Valley Authority through the Electric Farm and Home Authority, plans to finance the sale of electric ranges, refrigerators and water heaters in the TVA district. But to make this equipment eligible for financing by the EFHA, lower prices were held to be necessary

After setting up quality specifications for the appliances, the TVA received bids from fifteen manufacturers for special low-cost designs. Preliminary bids received were all too high, according to D. E. Lilienthal, director of the TVA. He indicated that much lower selling prices would be necessary—between $70 and $80 for the refrigerator, for example.

According to Mr. Lilienthal to meet these low prices, engineers must develop an entirely new type of appliance, designed to take advantage of lower manufacturing and assembly costs expected with larger quantities in production. It was not explained whether or not this would necessitate purely utilitarian equipment with perhaps a sacrifice in quality, nor was it shown how manufacturing costs could be reduced further in the face of rising costs brought about by the N.R.A. and other codes.

Group Versus Unit Drives Again to the Forefront

Reminiscent of the days of hot argument over unit versus group drives, a new organization operating under the name of Mechanical Power Engineering Associates, has begun an extensive investigation to establish a more scientific basis for selecting the type of drive most suitable for a particular application.

Reports of investigations made by these engineers cite numerous examples wherein the type of drive used was found to be not the most efficient available. Most of the instances relate to small single-motor driven machines. According to the findings of the Mechanical Power Engineering Associates, under certain conditions when a number of such machines are placed close together in the factory, it is more economical to use a group drive, all of the machines being driven by one motor through lineshafts and belts.

It is too early to determine whether the investigations reported by the Mechanical Power Engineering Associates refer only to isolated examples or whether they have general application. If the latter, it may become necessary for engineers to design their motor-driven machines so that they can be converted economically to belt drive. This of course would only apply to the relatively small machines driven by a single motor.

Future activities of the Mechanical Power Engineering Associates will be watched with much interest because of the possible effect that their findings may have upon the future design of motor-driven machines.

Impact Tests On Cast Iron Found Unsatisfactory

Impact testing is not a satisfactory means of specifying the properties of cast iron according to a report of Committee A-3 of the American Society for Testing Materials. Unreliable results were obtained using small impact specimens such as are used for steel. All notched specimens were unsatisfactory, and tensile impact tests did not give a reliable index of impact value.

This report concludes that the bending curve of the transverse test, if the test is accurately carried out, is capable of giving most of the information needed about cast iron. As a supplementary test the Brinell hardness determined over the whole cross section is the most valuable. A copy of this report, "Impact Testing of Cast Iron," can be obtained from the American Society for Testing Materials, Philadelphia, Pa. Price 50 cents.

MEETINGS

American Gear Manufacturers Association—Annual meeting, Penn-Lincoln Hotel, Wilkinsburg, Pa., May 3-5. J. C. McQuiston, secretary, Penn-Lincoln Hotel, Wilkinsburg, Pa.

National Petroleum Meeting — A.S.M.E. Petroleum Division National Meeting, Tulsa Hotel, Tulsa, Okla. May 14-17. Harold Adkinson, secretary, Tulsa Bldg., Tulsa, Okla.

Society of Automotive Engineers —Summer meeting, Saranac Inn, Lake Saranac, N. Y., June 17-22. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Institute of Electrical Engineers — Summer meeting, Homestead Hotel, Hot Springs, Va., June 25-29. H. H. Henline, secretary, 33 West 39th St., New York, N. Y.

American Society of Mechanical Engineers — Semi-annual meeting, Cosmopolitan Hotel, Denver, Colo., June 25-29. Calvin W. Rice, secretary, 29 West 39th St., New York, N. Y.

Expansion Coefficients of Bearing Bronzes

Report of a recent investigation by the Bureau of Standards states that the addition of tin to copper alloys, or lead to copper-tin alloys increases the coefficients of expansion. Expansion curves taken during the first heating were found to indicate higher coefficients than obtained in the first cooling, while expansion effects during the second heating were generally higher than those found in the first heating.

Equations given in the report show for various compositions the relationship between expansion coefficients and lead or tin content for temperatures ranging from room temperature up to 200 deg. C. By the use of these equations, and curves accompanying them, coefficients of expansion can be predicted for other cast copper-tin and copper-tin-lead alloys. In the March number of the *Bureau of Standards Journal of Research* is given a complete report of this work.

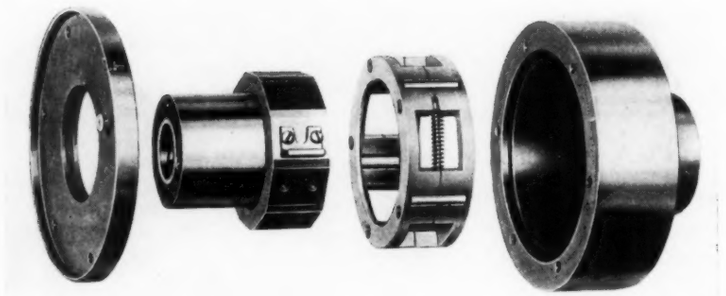
Aviation Making New Records

Progress in aeronautical design is strikingly reflected in the schedules of the commercial airlines as reported by TWA. Serving 178 cities, commercial airplanes now average 158,537 miles per day, 40 per cent of the flying being done at night. Speed records are also being broken. Leaving Los Angeles with fourteen passengers, one of the new Douglas Airliners landed at Newark Airport 13 hours and 5 minutes later, having flown the distance at an average speed of 203 miles per hour, much of the time at 18,000 ft. altitude.

NEW MATERIALS and PARTS

Hilliard Over-Running Clutch

Designed for instant engagement and release, clutch has a series of rollers running between an outer ring and a series of flat surfaces on an inner hub. Rollers are held in a bronze cage retained in position by a small bracket on the hub. Drive may be applied either through the outer sleeve or the inner hub, and direction of rotation can be readily reversed in the field. There are two flat surfaces on the hub for each roller, providing new wearing surfaces by moving the hub to a new position within the roller cage. Unit is said to be sufficiently flexible to serve either as a clutch or as a coupling to connect two shafts. Available in twelve sizes for capacities from $\frac{1}{4}$ to 100 hp., at 100 r.p.m. Also furnished with a trip arrangement for use as an automatic clutch or intermittent drive. The Hilliard Corp., Elmira, N. Y.



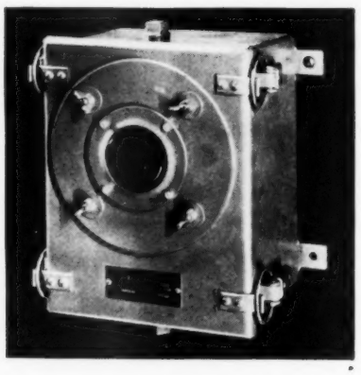
Acme Variable-Voltage Adjuster

For use with electrical products that may be connected to circuits other than 110 volts, or where an adjustable voltage is required. Device consists of a transformer, a voltmeter and an adjustment knob for selecting the desired voltage. Supplied with an 8-ft. attachment cord. Adjuster has a capacity of 100 to 150 volt-amperes, and is supplied with five voltage ranges for either 115-volt or 220-volt output.

Dimensions are 5 in. high, $3\frac{7}{8}$ in. wide, $4\frac{3}{8}$ in. deep. The Acme Electric & Manufacturing Co., 1444 Hamilton Ave., Cleveland, Ohio.

Light Source for Photocells

Consists of an industrial light-source lamp rated at 6 volts, 5 amp., and 3,000 hr. life, with a lens for condensing the light rays into a parallel beam. Provisions are included for adjusting the focus of the lens without removing the cover of the unit. Lamp current is supplied by a transformer inside the case. A light intensity of 24 ft. candles is obtained at 25 ft., the intensity varying inversely as the square of the distance for other distances. Said to be adapted to applications requiring long light beams, or where the light beam must penetrate steam, dust or semi-



translucent substances. Entire unit mounted in a steel cabinet for either indoor or outdoor mounting. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Jefferson Small Transformer

Inclosed within a one-piece housing, 110 volt transformer, has a secondary supplying 10 volts, 0.5 amp. Listed as standard by the Underwriters' Laboratories. Dimensions are $2\frac{1}{4}$x$2\frac{3}{8}$x$2\frac{3}{8}$ in. Claimed that electrical circuit is so designed that core loss will not register on the average watt-hour meter. Available for 110 or 220 volts, for 25 and 60 cycles. Jefferson Electric Co., Bellwood, Ill.

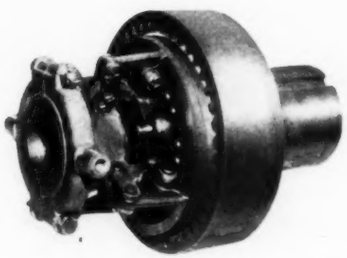
Marsh Combined Gage and Thermometer

For use on domestic heating systems or steam apparatus using low pressures. Instrument is constructed with two separate and independent sets of indicating mechanism, both sets operated by a bourdon tube. Temperature indicator operated by the liquid expansion principle, pressure mechanism of conventional type. Movement of the bourdon tubes is communicated to the indicating hands through a compensated movement mechanism. Available in a standard unit with a temperature scale of 80 to 250 deg. F., and a pressure scale of 0 to 30 lb. pressure, with an altitude reading from 0 to 70 ft. height of water. Available for other temperature and pressure ranges on special

order. Drawn case incloses a dial 3½ in. in diameter. Case available in black finish or in colors, or plated to match other apparatus. Jas. P. Marsh Corp., 2073 Southport Ave., Chicago, Ill.

Twin Disc Clutches

Available in eight sizes from 6½ in. to 24 in. in diameter, with 1, 2 or 3 plates. Spiders are available in three different styles for various installations, and are of the internal gear-tooth type. Clutches have toggle ac-



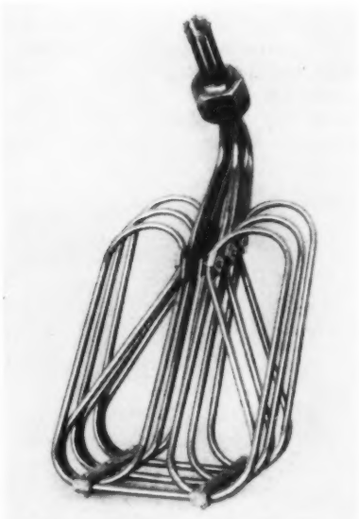
tion, and inner plates are gear-tooth driven from the hub. Twin Disc Clutch Co., 1328 Racine St., Racine, Wis.

Monobelt Leather Belting

Using a specially prepared vegetable tanned leather having high adhesion and stretched by a special process, internal friction is said to be reduced, giving a higher horsepower capacity. Made of waterproof leather put together with waterproof cement, belting is said to have more resistance to abnormal atmospheric conditions. Belting is available in graduated thicknesses, increasing with the width of the belt, only one thickness for each width. Double-ply belts are made of two plies possessing different degrees of elasticity, the outer ply being more elastic than the inner ply. Alexander Bros., Inc., 406 N. Third St., Philadelphia, Pa.

Two G.E. Heating Units

A portable copper-sheathed unit for heating water is formed in a self-supporting helix with vertical extensions ending in a sealed-plug socket. Since only the spiral is active, only 5 in. of water is necessary. The heater is rated at 1,300 watts at 115 or 230 volts. For use in heating linseed oil, the heater illustrated is made entirely of Monel metal, providing means for accurate



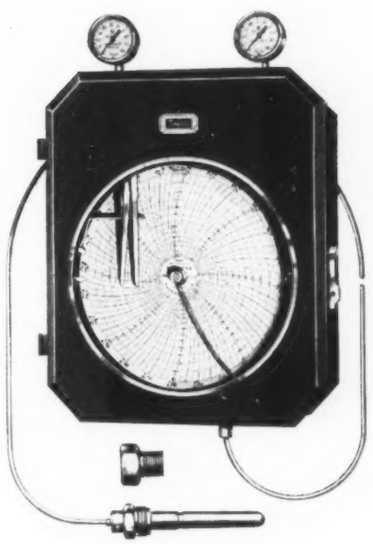
temperature control without discoloring the oil. Rated at 12 kw., 330 volts, d.c., or a.c. General Electric Co., Schenectady, N. Y.

Omega Blue Print Paper

Made of wood pulp, paper is said to fulfill all specifications for rag stock papers besides offering a price advantage. More uniform color is also claimed to result from the fact that the wood pulp paper is not sized and the photo-sensitive chemicals are therefore more evenly distributed in the paper stock. Made possible by recent developments in wood pulp processes, the new product is claimed to give a deep blue and pure white with exceptional speed. W. Bersch, 110 East 42d St., New York, N. Y.

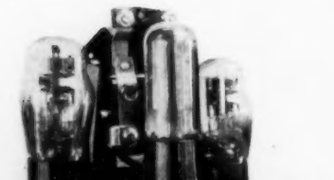
Taylor Recording Regulators

Incorporating a sliding adjustment over a graduated scale, regulator gives throttling or on-and-off diaphragm valve action, adjusting the sensitivity for balancing regulator performance against the time lag in the apparatus under control. The same regulator mechanism and means of adjustment is used in temperature, pressure, rate of flow, or liquid level controllers. The use of light weight materials in the mechanism is said to produce a higher degree of accuracy in combination with other refinements of design. Regulator action can be converted from direct to reverse without addition of parts, and regulators are adaptable to different temperature or pressure ranges by substituting a tube system and chart with the required range. Controller operates a diaphragm valve by regulating the flow of air in the control line, using a 25 lb. air pressure supply, and is compensated for pressure variations. Regulators are inclosed in a die-cast aluminum case supplied in two styles: for face-mounting, or flush-mounting on panel boards. Electric chart-driving mechanism is standard with a spring-driven mechanism optional. Taylor Instrument Companies, Rochester, N. Y.



Industrial Photo-Electric Relay

For industrial uses such as conveyor synchronizers, safety devices on machines, grading small parts, counting and register control, this light-operated relay is said to give up to 600 operations per minute. Sensitive relays are eliminated by the use of a new electrical circuit, the tubes operating a heavy-duty contactor directly. The only moving mechanical part is the armature of the contactor. Either gas or vacuum phototubes can be used, and phototubes can be mounted at a distance of 25 ft. from the amplifier, using the standard housings available. Relay is adaptable to an on or off operation of the contactor when the light beam is interrupted. Relay can also be operated by high resistance contacts or a microphone instead of a phototube. Complete device is mounted in a case 6⅛ in. square,



9⅛ in. high. Silver-to-silver wiping contacts on contactor are rated at 20 amp., 115 volts, a.c., non-inductive load; or 3 amp. 115 volts, d.c. Power supply required is 40 watts for any frequency from 25 to 60 cycles. Also available for d.c. power supply. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Paragon Time Control

Designated as Type G, is designed for use with thermostats, oil burner controls, machine controls and other electrical circuits up to 600 watts power. Movement contains but four cut-brass wheels and steel pinions, driven by a selt-starting synchronous motor with self-lubricated gears within a dust-proof die-cast housing. Dial is

direct-driven from motors without friction clutches. Fast-make and fast-break switch is constructed of bakelite and bronze with silver contacts. Available in single-throw single-pole, double-throw, or special two-circuit types. Pressed steel case is 8 in. high, 4 in. wide, 3 in. deep, has six replaceable knockouts. Finished in black crackle lacquer. Paragon Electric Co., 409 S. Dearborn St., Chicago, Ill.

Explosion-Proof Flexible Conduit Fitting

Watertight and explosion-proof, this flexible fitting is said to have all the mechanical features of standard flexible conduits. Designed for use on electrical equipment for filling stations, refineries, grain elevators and other applications where hazardous conditions exist. Also suitable for breweries, distilleries and chemical plants, providing a watertight electri-

cal installation. Fitting is said to withstand 500 lb. hydrostatic pressure. Each fitting is tested against leaks at 50 lb. air pressure under water. Listed by Underwriters' Laboratories for use in Class I, Group D, and Class II, Group G, hazardous locations. The Pyle-National Co., 1334-58 North Kostner Ave., Chicago, Ill.

Permite Aluminum Paint

Manufactured by a new vacuum process, this paint is said to spread more easily and to produce a more uniform brilliance. Claimed that there are more separate and individual particles of aluminum powder in this ready mixed paint than are found in hand-mixed paints, since individual powder particles are separated. Paint Division of Aluminum Industries, Inc., Cincinnati, Ohio.

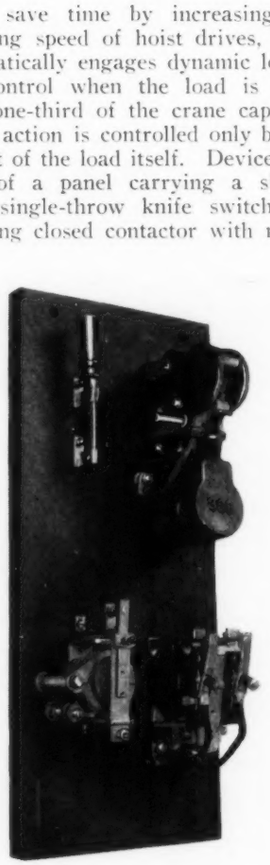
Water-Cooled Wheel Head

For precision grinding equipment, is said to be capable of continuous operation without appreciable rise in bearing temperature. Oil chamber and bearings are surrounded by a cooling jacket through which the ordinary grinding coolant is continuously pumped. Spindle construction is said to eliminate chatter. Precision ball

bearings run in oil, and the spindle is designed to rest within the bed of the cross slide without overhang. Available in three sizes with maximum spindle speeds of 10,000, 15,000 to 18,000, and 25,000 r.p.m.; furnished with either quill or solid spindle. The Wicaco Machine Corp., Wayne Junction, Philadelphia, Pa.

High-Speed Lowering Relay for Cranes

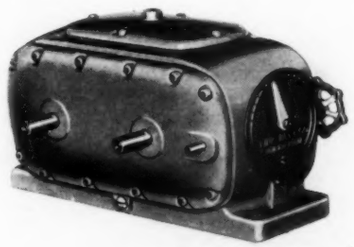
To save time by increasing the lowering speed of hoist drives, relay automatically engages dynamic lowering control when the load is more than one-third of the crane capacity. Relay action is controlled only by the weight of the load itself. Device consists of a panel carrying a single-pole, single-throw knife switch and a spring closed contactor with neces-



sary relays. Can be applied to any d.c. motor-operated crane equipped with series motors, requiring little alteration in the main controller and crane wiring. Available for any size hoist motor, for circuits from 110 to 550 volts. Harnischfeger Corp., Milwaukee, Wis.

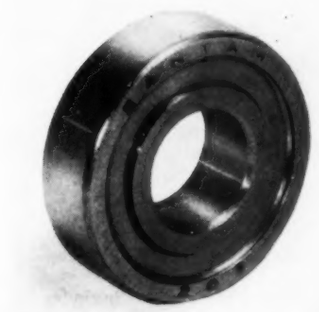
Link-Belt Variable-Speed Transmission

For fractional horsepower duty, self-adjusting, positive variable-speed transmission using a roller chain drive. Maximum ratio of speed variation is 10 to 1. Totally inclosed device is of all-metal construction, self-lubricated in an oil bath. Chain used for transmitting power is a side-contact roller type, consisting of steel links connected by hardened steel pins in hardened steel bushings. Openings in each pitch of links provide pockets for a hard steel roller on each side of the chain, the rollers protruding sufficiently to roll in contact with a hardened steel conical disk forming the driver and driven wheels. Spring pressure applied to the two disks of one wheel is said to maintain correct tension and compensate for chain wear. Transmission is provided with a speed ratio indicator. Made in horizontal or vertical types, with or without reduction gearing, and can also be supplied with a built-in motor. Link-Belt Co., 910 S. Michigan Ave., Chicago, Ill.



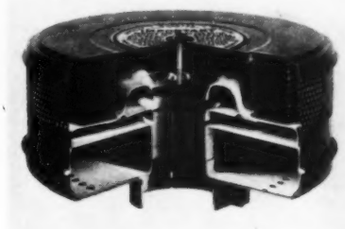
Shielded Ball Bearings

Precision radial bearings conforming to S.A.E. No. 203 embody a shielded construction to retain the lubricant and prevent dust or dirt from entering the bearing. For applications where abrasive foreign matter is present. Also, adaptable to materials handling applications, textile equipment and food machinery to prevent damage by oil from bearings. The Bantam Ball Bearing Co., South Bend, Indiana.



Burgess Intake Silencer

Built on the same principle as a straight-through muffler, silencer and air-cleaner combined uses cellulose fiber saturated with oil. Capillary action of dust and dirt as it accumulates is said to draw oil out of fibers, maintaining efficiency as an air cleaner. Silencer chamber is designed to trap and absorb intake noises. Available for internal combustion engines for automotive and industrial use. Acoustic Division of the Burgess Battery Co., Madison, Wis.



Metzgar Light-Duty Caster Wheels

Built up of a number of wedges cut from selected maple, caster wheel is constructed so that wearing surfaces are end-grain. Maple wedges are assembled by a gluing process said to result in a waterproof and weather-proof wheel. Tread surface of wear-resisting end-grain is claimed to protect floors, roll easily and absorb shock. Fitted with a ball bearing, wheel is available in diameters from 2½ in. to 10 in. in any face thickness, hub length and bore. Metzgar Co., Inc., Grand Rapids, Mich.

Combination Magnetic Clutch and Brake

With an outside diameter of 5 in., brake has a torque rating of 40 in.-lb., and the clutch rating is 8 in.-lb. Device can be obtained for a maximum bore of ⅞ in., and can be operated from any d.c. supply up to and including 250 volts. Brake is spring actuated, assuring a positive stop if power supply is interrupted. Magnetic Manufacturing Co., Milwaukee, Wis.

Sentinel Breakers

For the protection of fractional horsepower motors, this disconnecting switch permits sufficient time delay to start small motors having high starting current, but disconnects motors from the line when stalled. Designed to protect against burn-outs caused by low voltage or overload conditions, in household appliances, small tools, fans and blowers. Self-protecting element is a combination of bi-metal and auxiliary heater, is trip-free and cannot be held in the "on" position, while an overload exists. Quick-make and quick-break mechanism in breaker is completely inclosed in a molded case, with all metal parts made of non-corrosive metals or cadmium plated. Thermal heaters available in 19 ratings, ranging from 1.4 amp. to 14.2 amp., for protecting motors with full load ratings from 1.1 amp. and 12.9 amp. Westinghouse Electric & Mfg. Co., Pittsburgh, Pa.

NEW BOOKS and PUBLICATIONS

Metallurgy of Iron and Steel

Bradley Stoughton. 559 pages, 6x9 in. Red clothboard covers. Published by McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $4.

For the fourth edition of this book an entirely new manuscript was prepared, with material rearranged more in a historical sequence, and with less space devoted to subjects already covered in other textbooks.

Early history and a summary of iron and steel manufacture are given in the first three chapters. The next seven chapters, making up about one-half of the text, cover ore smelting, steel making and mechanical treatment —subjects more of interest to steel makers than designers. Two chapters on the constitution and heat-treatment of steel and two on cast iron contain much fundamental data bearing directly on the design problem of selecting materials. A closing chapter on alloy steel and alloy cast iron stresses physical properties and applications rather than steel making processes, and should be of real assistance to every engineer.

Chapter XI in 30 pages includes an unusually clear explanation of the iron-carbon diagram. This chapter is an excellent non-technical yet accurate treatment of this important engineering subject. In fact, throughout the text the author has refrained from using the technical terminology found in so many of the texts on this subject.

Vocational Guidance in Engineering Lines

Publication sponsored by the American Association of Engineers, 521 pages, 6x9. Green clothboard covers. Published by the Mack Printing Co., Easton, Pa. Price $2.50.

Although prepared for the purpose of serving as a practical yardstick to measure the natural fitness of young men aspiring to be engineers, this compilation includes much of immediate interest to the practicing engineer, especially the recent graduate. Prepared under the direction of J. A. L. Waddell, chairman of the editorial committee, the text is made up of 60 chapters written by more than 50 recognized engineers.

Following a somewhat verbose editorial foreword and introduction of 90 pages, nine chapters, each prepared by a different author, present an outline of the field of various branches of engineering. Almost every chapter includes a brief history of the branch of engineering and a description of the field with which it deals, as well as special demands and the practical aspects of pay and advancement. Several specialties in each of the fields of engineering are covered in 40 short chapters, making up approximately one-half of the text. These short chapters, each written by an authority, show the character, special requirements and opportunities in each specialty.

▼

Ball Bearings—New Departure Mfg. Co., Bristol, Conn. Booklet, 12 pages, "Bearings That Simplify Design," illustrates applications of automobile bearings specially designed for front and rear wheels, and differential.

Electrical Control — The Trumbull Electric Mfg. Co., Plainville, Conn. Catalog 16-A, 8½x10 in., 48 pages, "Trumbull Ready-Reference Catalog," lists in tabular form various types of safety switches, motor controllers, and other electrical equipment, such as knife switches, circuit breakers, panel boards. Four pages are devoted to wiring data for electric motors.

Electrical Heaters—General Electric Co., Schenectady, N. Y. Catalog GEA-1520-B, 52 pages, 8x11 in. "Electric Heating Units and Devices," includes immersion heaters, cartridge units, strip heaters, heating devices and control equipment.

Electric Motors—The Holtzer-Cabot Electric Co., 125 Amory St., Boston, Mass. Two folders on motors for heating and ventilating fans include data on applications, types and controls for propeller and centrifugal fan drives.

Electric Motors — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Leaflet, 4 pages, "Quiet Operating Motors," showing distinctive features and construction for induction motors designed for quiet operation.

Electronic Equipment — General Radio Co., 30 State St., Cambridge, Mass. Catalog G, Part II, 48 pages, with specifications and catalog information on stroboscopes, oscillographs, transformers, frequency meters, standards, and power radio equipment.

Finishes — Hilo Varnish Corp., 42 Stewart Ave., Brooklyn, N. Y. Bulletin No. 8, four pages, "Hilo Crystal Finishes," and Bulletin No. 16, eight pages, "Vehicles for Aluminum Paint." Data on industrial applications, types available and finishing processes.

Flexible Couplings—Poole Foundry & Machine Co., Baltimore, Md. Catalog 34, 6x9 in., 72 pages, "Flexible Couplings," including engineering data on horsepower ratings, maximum speeds, dimensions, and weights for several types of standard and special designs. Typical applications and principles of operation are also given.

Gears—Horsburgh & Scott Co., 5114 Hamilton Axe., Cleveland, Ohio. Catalog No. 34, 4½x6 in., 448 pages, "Horsburgh & Scott Gears and Speed Reducers," lists all types of gears, giving catalog number and prices. For spur gears both circular pitch and diametral pitch are listed for nearly all numbers of teeth; tooth loads and horsepower ratings are included for each standard gear. Speed reducer section comprising 120 pages gives torque ratings, dimensions and details of construction for worm and herringbone types.

Mica Insulation — Continental-Diamond Fibre Co., Bridgeport, Pa. Booklet, 6x9 in., 24 pages, "Micabond," giving electrical insulation properties, sizes and specifications for processed mica plates, sheets, paper cloth, tubing and v-rings.

Nickel-Chromium Alloy—American Manganese Steel Co., 6600 Ridge Ave., St. Louis, Mo. Folder giving typical applications and characteristics of six nickel-chromium alloys for resisting heat, corrosion or acids.

Photocells — Acousto-Lite Corp., Ltd., 2908 S. Vermont Ave., Los Angeles, Calif. Booklet, 5x7 in., 30 pages, "Manual of Selenium Cells and Circuits," includes data on illumination, and a description of the Acousto-Lite Selenium Cell. Properties, intensity limits, and the uses of low and high resistance selenium cells are illustrated with tables, charts, and wiring diagrams.

Plastics—Bakelite Corp., 247 Park Ave., New York, N. Y. Sixth edition of "Bakelite Molded," is a booklet, 6x9 in., 48 pages, covering various molding materials, their uses, and a brief description of the molding process and necessary equipment. Applications are described and a variety of products are attractively illustrated. Physical properties are given in tables for wood-flour, mineral-filled and high-impact material. Designs that should be avoided are discussed.

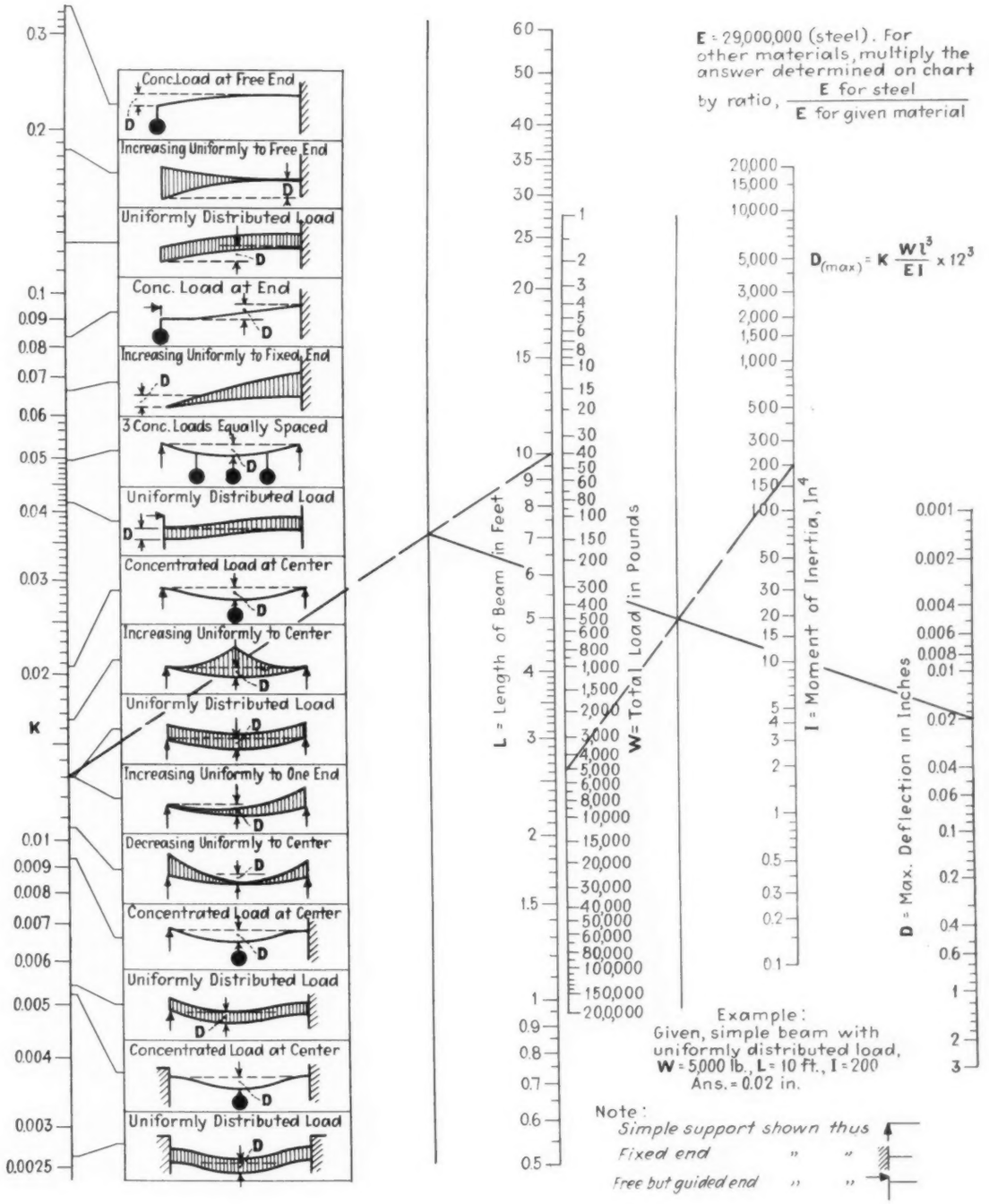
Pumps—Roots-Connersville Blower Corp., Connersville, Ind. Bulletin 60-B10, 4 pages, gives applications and capacities for cycloidal rotary pumps.

Shafting—Union Drawn Steel Co., Massillon, Ohio. Booklet, 16 pages, "Union Cold Finished Shafting," describes the process, manufacturing standards, physical properties and surface finish of shafting finished by drawing through dies.

Shims—Laminated Shim Co., Inc., 2124 Forty-Fourth Ave., Long Island City, N. Y. Booklet, 8 pages, "The Solid Shim that Peels," includes types, typical uses, and stock thicknesses.

Deflection of Variously Loaded Beams

E. COWAN


Conc. Load at Free End
Increasing Uniformly to Free End
Uniformly Distributed Load
Conc. Load at End
Increasing Uniformly to Fixed End
3 Conc. Loads Equally Spaced
Uniformly Distributed Load
Concentrated Load at Center
Increasing Uniformly to Center
Uniformly Distributed Load
Increasing Uniformly to One End
Decreasing Uniformly to Center
Concentrated Load at Center
Uniformly Distributed Load
Concentrated Load at Center
Uniformly Distributed Load
K
D
L = Length of Beam in Feet
W = Total Load in Pounds
I = Moment of Inertia, In⁴
D = Max. Deflection in Inches
E = 29,000,000 (steel). For other materials, multiply the answer determined on chart by ratio, E for steel / E for given material
$D_{(max)} = K \frac{Wl^3}{EI} \times 12^3$
Example:
Given, simple beam with uniformly distributed load,
W = 5,000 lb., L = 10 ft., I = 200
Ans. = 0.02 in.
Note:
Simple support shown thus
Fixed end " "
Free but guided end " "

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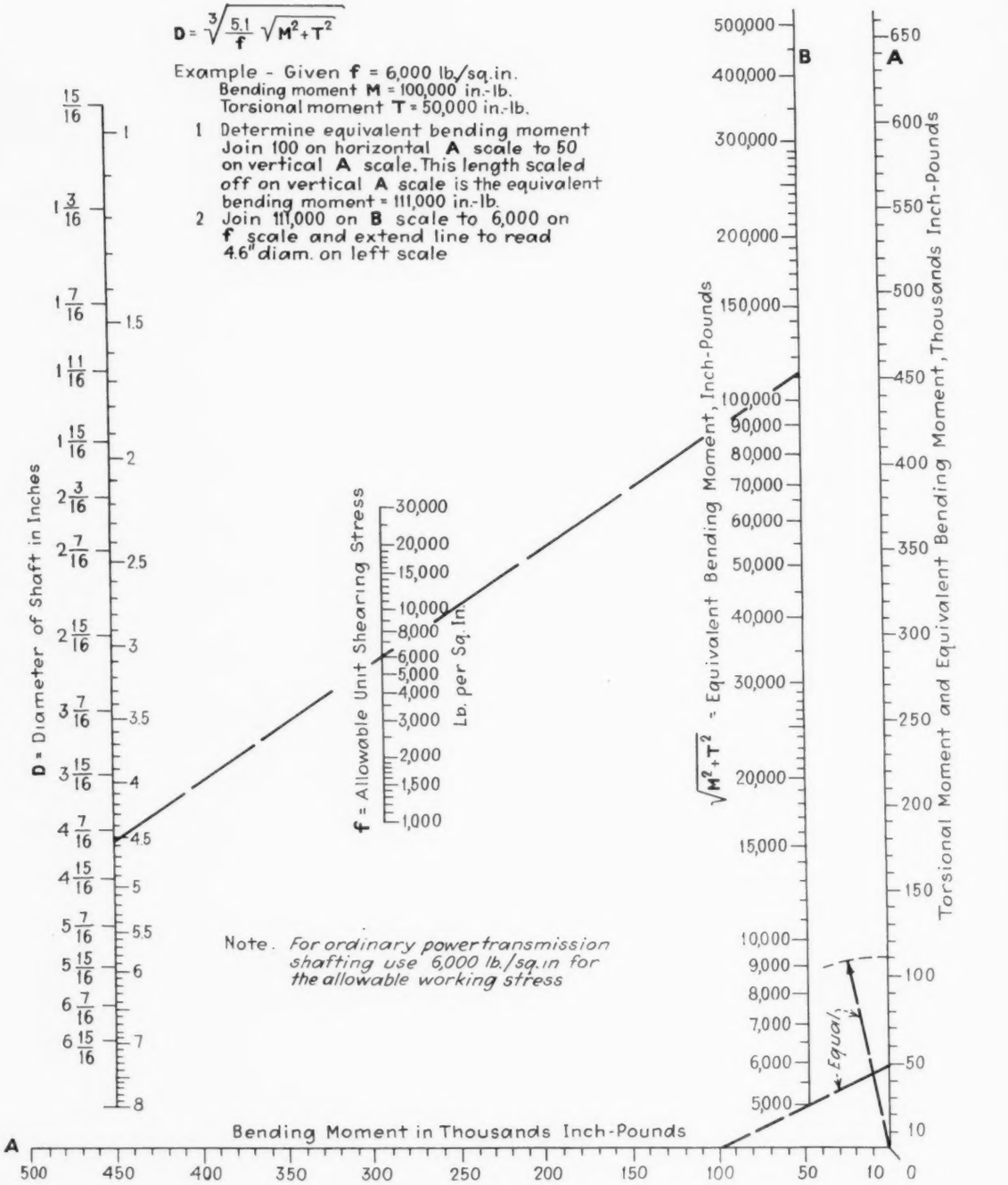
Shaft Diameters for Torsion and Bending

E. COWAN


$D = \sqrt[3]{\frac{5.1}{f}\sqrt{M^2+T^2}}$
Example - Given f = 6,000 lb./sq.in.
Bending moment M = 100,000 in.-lb.
Torsional moment T = 50,000 in.-lb.
1 Determine equivalent bending moment Join 100 on horizontal A scale to 50 on vertical A scale. This length scaled off on vertical A scale is the equivalent bending moment = 111,000 in.-lb.
2 Join 111,000 on B scale to 6,000 on f scale and extend line to read 4.6" diam. on left scale
D = Diameter of Shaft in Inches
f = Allowable Unit Shearing Stress Lb. per Sq. In.
$\sqrt{M^2+T^2}$ = Equivalent Bending Moment, Inch-Pounds
Torsional Moment and Equivalent Bending Moment, Thousands Inch-Pounds
Bending Moment in Thousands Inch-Pounds
Note. For ordinary power transmission shafting use 6,000 lb./sq.in for the allowable working stress
Equal

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